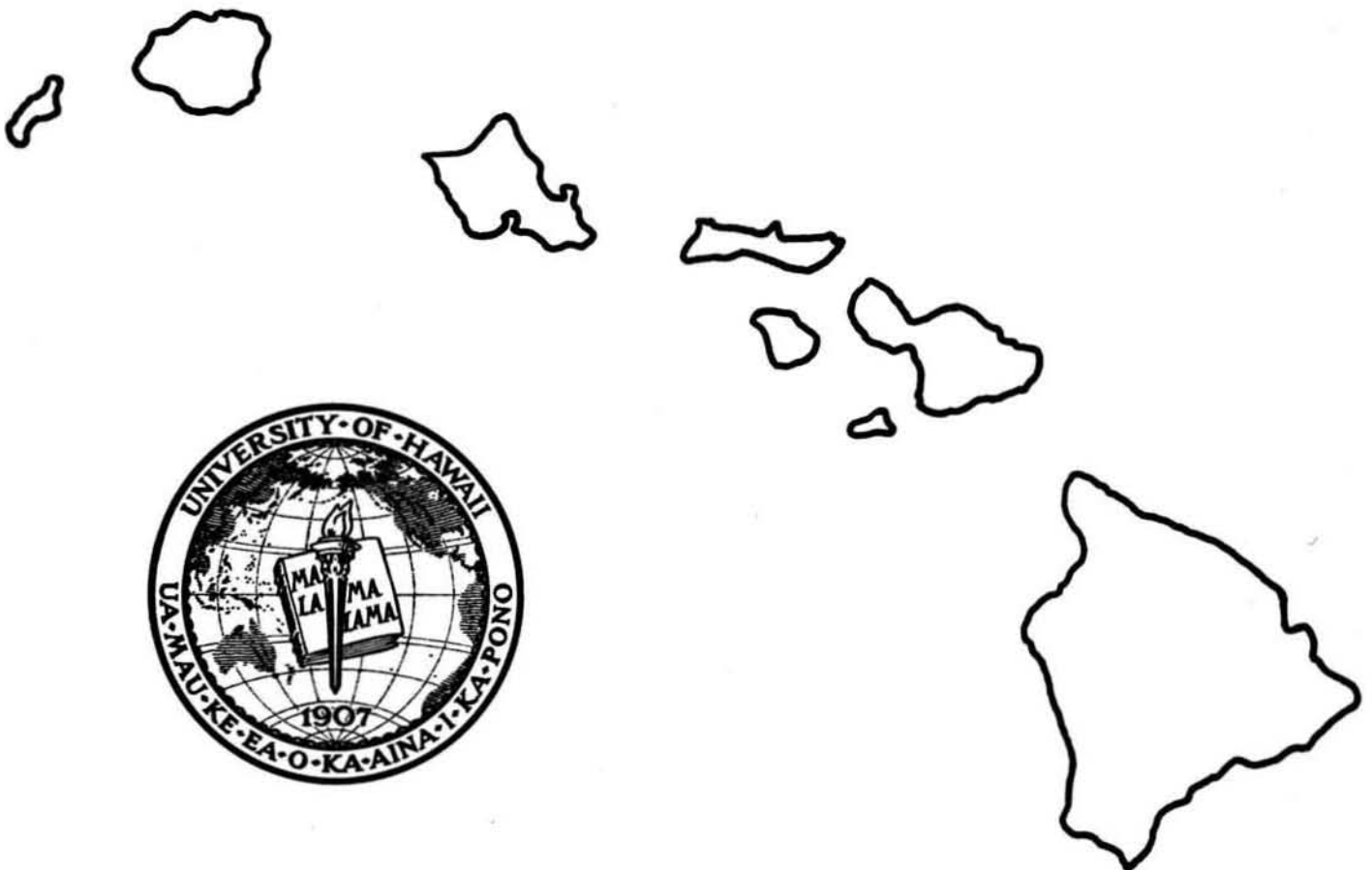


THE
**HAWAII GEOTHERMAL
PROJECT**

CHARACTERISTICS OF
VAPOR FLASHING GEOTHERMAL PLANTS

TECHNICAL REPORT NO. 6
November 15, 1974



HAWAII GEOTHERMAL PROJECT
ENGINEERING PROGRAM

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November 15, 1974

Prepared Under
NATIONAL SCIENCE FOUNDATION
RESEARCH GRANT NO. GI-38319

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ABSTRACT

A study of the important parameters of a vapor flashing plant and characteristics of its major components is presented. The investigations by others showed that a steam and water mixture can be transported in a single pipe without the problems of water hammer, cavitation, and vibration, and that the pressure drop of a two-phase fluid can be determined using Lockhart and Martinelli correlation. The optimum flashing temperature has been calculated by numerical method, and the effects of wellhead pressure and number of stages on power output were analyzed. The design and operating features of a bottom outlet cyclone separator are given. The steam rates of a mixed pressure turbine for geothermal application were estimated in terms of the size, loading, throttle pressure and temperature, and condensing pressure. A procedure is described to determine the performance and size of an induced-draft, cross-flow cooling tower. By performing a simple heat balance calculation, effects of non-condensable gas content and condensing pressure on the net power output were determined. Finally, the possible arrangements of the heat rejection equipment and the factors involved in the selection of a plant site are discussed.

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CHAPTER I

INTRODUCTION

Geothermal deposits are classified as vapor dominated, liquid dominated, or dry rock. The existing geothermal plants over the world exploit the first two types of geothermal reservoirs. Recently, methods have been proposed to extract energy from the earth's crust containing essentially hot dry rock. The method of converting the geothermal energy to electric power depends upon the type of geothermal reservoir and the quality of geothermal fluid. The liquid dominated fields are believed to be many times more abundant than the vapor dominated fields, yet the number of existing plants exploiting the two kinds of reservoirs are almost equal. Four kinds of plants have been proposed to harness energy from the liquid dominated fields: binary vapor-turbine plant, plant using secondary steam in a closed cycle, plant based on total flow concept, and vapor flashing plant. In a binary vapor-turbine plant, working fluid like isobutane or Freon extracts heat from the geothermal fluid and operates in a Rankine cycle to produce power. In a plant using secondary steam, pure water instead of isobutane or other refrigerants is used as the working fluid of the cycle. As to the total flow concept, the thermal energy of hot brine is converted to kinetic energy by expansion of the liquid-vapor mixture through a converging-diverging nozzle, and the kinetic energy of fluid is transformed to mechanical energy through an impulse turbine. This study deals only with the vapor flashing plant, in which the primary steam from the well and the secondary steam by flashing are separated from liquid water and then fed to a turbine. The exhaust vapor is usually condensed in a direct contact barometric condenser and the non-condensable gases are removed by gas extractors. A cooling tower is often used to recycle the cooling water which removes heat from the exhaust vapor. As the liquid is brought out of the reservoir, its pressure is reduced, and a part of the liquid water becomes steam in the well. The liquid-vapor mixture from the well contains large amounts of dissolved salt and non-condensable gases. The impurities in water cause the problems of scaling and corrosion. The non-condensable gases accumulate in the condenser and they must be removed by gas-extractors to maintain adequate condensing pressure. As compared to

a conventional power plant, the throttle pressure of turbine in a vapor flashing plant is relatively low. The optimum operating pressure depends upon factors such as the pressure and temperature of fluid in reservoir and the performance of the well. The objective of this study is to recognize the important parameters of a vapor flashing plant and to study the characteristics of its major components.

In Chapter II, the feasibility of transporting steam-water mixture in a single pipe is analyzed. The optimum flashing and wellhead pressures are discussed in Chapters III and IV. Chapter V deals with the development of an efficient steam-water separator. The important considerations in the design of a mixed pressure turbine for geothermal applications are given in Chapter VI. Chapter VII is concerned with the study of heat rejection equipment which consists of barometric condenser, gas extractor, and cooling tower. A heat balance of a vapor flashing plant using two-stage steam ejector is included to show how optimum condensing pressure may be determined. Finally, the possible arrangements of the heat rejection equipment and the factors involved in selecting a plant site are presented.

CHAPTER II

PRESSURE DROP OF TWO PHASE FLOW IN PIPE

If the geothermal fluid flows out freely through a well in a liquid-dominated field, a part of the liquid will become vapor because of the pressure drop caused by friction and the potential energy change. The amount of vapor separated depends on the heat content of the fluid and the pressure drop. There are two options to treat the two-phase fluid at the wellhead. The first is to separate steam from brine at the wellhead, using a centrifugal cyclone separator. Thereafter, steam and brine are piped separately to the power plant where the flashed steam is produced from the brine at a lowered pressure, and is fed to the turbine in an intermediate-stage. This option was adopted at Wairakei under the fear of water-hammer, cavitation, surging flow and excessive vibrations which might occur in the transportation of two-phase fluid through a pipe. To prevent flashing, the hot brine was pressurized by pumps and cooled by the injection of cold water. The second

option is to transport the liquid-vapor mixture from the wellhead to the plant through a single pipe. The separation of vapor from liquid takes place in the plant. The advantages of the second option are the reduction in pipeline cost and the increase of power output.

Through a series of experiments James [27] and Takahashi, et al. [42] demonstrated that there was no serious difficulty, previously suspected, in the transportation of steam-water mixture. From experimental results and analytical considerations of flow patterns, they concluded that when the flow pattern is in the annular flow region, the flow is stable even in pipes of large diameters. Even flow regimes described as slug and stratified are not dangerous for a pipe filled with flowing water, since a slug or wave will not be able to attain high velocities with water and steam in its path. Cavitation cannot be a problem for a pipeline carrying steam-water mixture because of the presence of water vapor, carbon dioxide, hydrogen sulfide and other non-condensable gases. These gaseous compounds provide a buffer to the impact forces [30]. However, care should be taken in starting up and the line should be initially heated and pressurized with steam to avoid pressure fluctuations and local water accelerations. There has been only one incident of water hammer at Wairakei when steam entered a closed pipe containing a large quantity of cold condensate. In James' experiment, a mixture of flashing steam and water flowed into a cold pipe which contained cold water in the upper leg of expansion bend; no water hammer was detected and only slight vibrations were noticed. The conclusion of the experiment was that the transportation of steam-water mixture in pipes can be as acceptable as that of steam alone and is perhaps preferable when there is residual water in the line.

There are three methods being suggested for calculating the pressure drop of two phase flow. The first method, introduced by Benjamin and Miller [9], involves the direct solution of the continuity and energy equations. They assumed isentropic expansion for determining the density of the liquid water mixture. Although the expansion of the fluid is not isentropic due to the friction loss, the results of their calculations fit the experimental data very well, due to the fact that the density of the mixture varies in nearly the same manner whether an isentropic or isenthalpic expansion is assumed. For clean steel pipes, they found the friction factor to be .012.

The second method was initiated by James [27] who suggested that the pressure drop ΔP of a two-phase flow can be determined by first calculating the pressure drop ΔP_{sf} of steam flow in the mixture alone and then dividing it by $x^{0.5}$,

$$\Delta P = \Delta P_{sf} / x^{0.5} \quad (2.1)$$

where x is the dryness fraction. This equation was for moderate flows through the pipe, where the speed of the steam fraction is of the same order of normally recommended speed of steam alone. The calculated results by this equation were found to be inconsistent with the experimental data. The third method was originated by Lockhart and Martinelli. James [26] and Takahashi, et al. [42] found that Lockhart and Martinelli's correlation gave calculated values of pressure drop in excellent agreement with experimental results for horizontal loops and that the pressure drop for vertical expansion loop could also be predicted quite accurately with an additional factor. The first step is to calculate the dimensionless number χ as follows:

$$\chi = \frac{\Delta P_{lf}}{\Delta P_{sf}} = \sqrt{\left(\frac{W_{lf}}{W_{sf}}\right)^{1.8} \cdot \left(\frac{\rho_{sf}}{\rho_{lf}}\right) \cdot \left(\frac{\mu_{lf}}{\mu_{sf}}\right)^{0.2}} \quad (2.2)$$

where ρ , μ , P and W are the density, viscosity, pressure and mass flow rates, and the subscripts lf and sf refer to the liquid and steam fractions respectively. Figure 1 gives the non-dimensional pressure loss factor ϕ as a function of χ and the total pressure drop is

$$\Delta P = \phi_{sf}^2 \Delta P_{sf} = \phi_{lf}^2 \Delta P_{lf} \quad (2.3)$$

James stated that the Lockhart and Martinelli correlation is applicable to the flows with steam fraction velocity not less than about 60 ft/sec. At lower velocities, slug flow appears and the correlation becomes less accurate. Pressure drop increases exponentially with the increase of velocity. The optimum pipe diameter D for reasonable pressure drop was recommended by him as follows:

$$D = 0.1365 \left(\frac{W \cdot x}{P^{0.47}} \right)^{0.5} \quad (2.4)$$

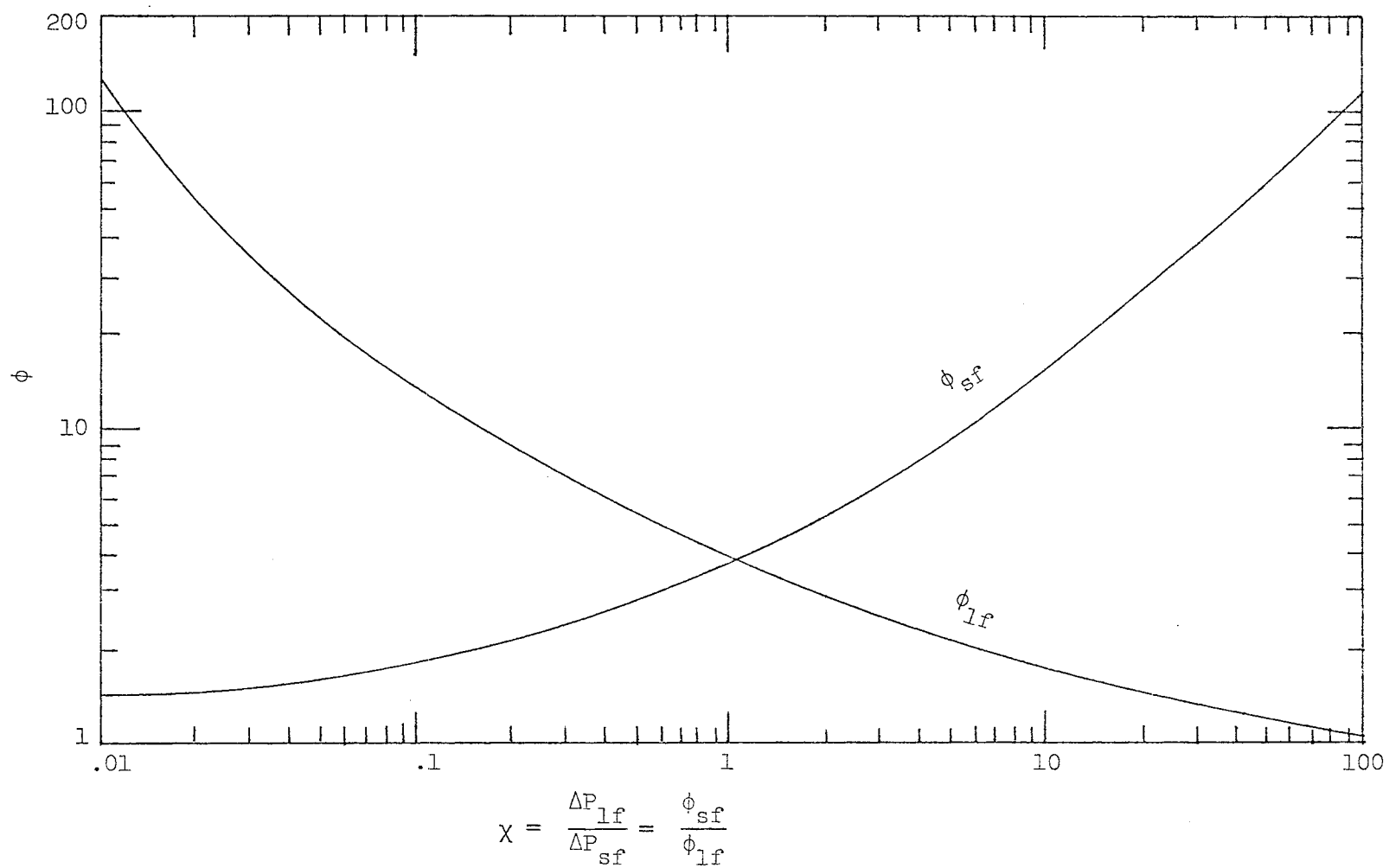


Figure 1. Lockhart Martinelli Correlation for Turbulent Liquid, Turbulent Vapor Case

Equation (2.4) can merely serve as a very rough guide. The optimum size of pipe can be determined only after the cost of piping and cost of power loss due to pressure drop are carefully appraised.

Total annual piping cost = annual capital cost of piping + annual cost of power loss. The annual capital cost of piping is a function of the diameter D of pipe; the annual cost of power loss is affected by the pressure drop, which is inversely proportional to the fifth power of diameter.

$$\Delta P = \phi_{1f}^2 \cdot f \cdot \frac{L}{D} \cdot \left[\frac{(1-x) \cdot W}{ID^2 \cdot \rho_{1f}} \right]^2 \cdot \frac{1}{2gc} \cdot \rho_{1f} = C \cdot \frac{(1-x)^2 \cdot W^2}{D^5 \cdot \rho_{1f}} \quad (2.5)$$

In summary, investigations by Takahashi [42] and James [27,30] have shown that two-phase geothermal fluid can be transported through pipes without suspected troubles of water hammer, cavitation and excessive vibrations. The Lockhart and Martinelli correlation is recommended for the determination of pressure drop.

CHAPTER III

FLASHING PRESSURES

In a vapor flashing system hot water is flashed to steam at a pressure lower than the wellhead pressure. The vapors thus generated may be used to drive a mixed pressure turbine. Flashing may be done in a centrifugal cyclone separator. Bengma [8] discusses the development and selection of a steam water separator. Chierici [12] draws some important conclusions about the feasibility of using a condensing plant.

Work output from the turbine is the product of steam flow rate and the available energy. In a simple flashing system, the lower the flashing pressure, the higher the steam production rate. However, available energy associated with each pound of steam decreases with lowering of the separator pressure. Therefore, there exists an optimum flashing pressure for obtaining maximum power from the hot water. At a pressure higher than the optimum, work output is small due to small available energy. At a pressure lower than the optimum, low turbine work results because of the diminished steam flow rate.

By employing multi-stage flashing, the temperature of discharged water is lowered and thus the work output of the plant can be greatly increased. The number of stages of flashing is a matter of economic justification. Power contribution of an additional stage decreases as the number of stages increases, and specific volume of steam increases rapidly at low pressure to cause high cost of the multi-stage arrangement. An advantage of multiple admission of steam to a mixed pressure turbine is to improve quality of the exhaust steam thereby reducing erosion of the turbine blades. Also steam flashed from separated water is practically free of gases. The percentage of gas content in the exhaust steam can be lowered by blending of the flash steam with the steam directly from the well. In a mixed-pressure turbine, there are separate inlets for steam at different pressures, and each inlet is equipped with its own control valve.

Hansen [22] stated that for a simple flashing cycle, optimum flashing pressure corresponds to a saturation temperature halfway between the well water temperature and saturation temperature of the condensate in the condenser. For example, for 400°F well water and 120°F condensate, the optimum flash temperature should be 260°F. He extended the rule to the multi-stage flashing plant. For well water at 400°F, condensate at 120°F, and three-stage flashing, optimum intermediate pressures should correspond to saturation temperatures of 330°F, 260°F and 190°F respectively. A procedure, described in the next paragraph, has been worked out to determine the exact optimum flash pressure with numerical methods and the results confirm Hansen's rule of approximation.

Using a three stage plant as an example, as indicated in Figure 2, we have the following equations from the consideration of mass and heat balance:

$$m_1 + m_4 = 1, \quad (3.1)$$

$$m_2 + m_5 = m_4, \quad (3.2)$$

$$m_3 + m_6 = m_5, \quad (3.3)$$

$$m_1 h_1 + m_4 h_4 + q_1 = h_0, \quad (3.4)$$

$$m_2 h_2 + m_5 h_5 + q_2 = m_4 h_4, \quad (3.5)$$

$$m_3 h_3 + m_6 h_6 + q_3 = m_5 h_5, \quad (3.6)$$

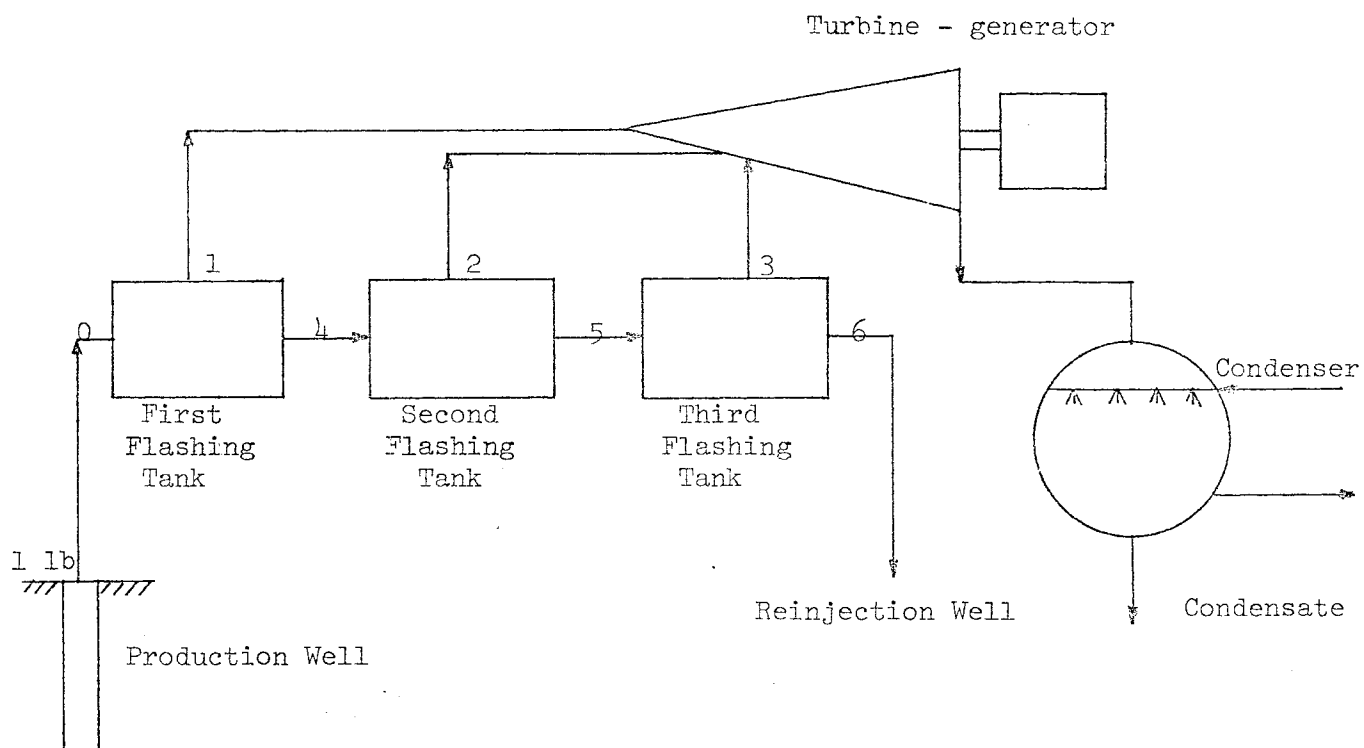


Figure 2. Three Stage Vapor Flashing Plant.

where q is the heat loss from the flashing tank per pound of fluid at wellhead, h is the specific enthalpy, m is the mass of the fluid, and the subscripts refer to the locations of points given in Figure 2. The conditions of steam and liquid water are not exactly saturated. Assume:

$$h_1 = hf_1 + [1 - x(1)] hfg_1, \quad (3.7)$$

$$h_2 = hf_2 + [1 - x(2)] hfg_2, \quad (3.8)$$

$$h_3 = hf_3 + [1 - x(3)] hfg_3, \quad (3.9)$$

$$h_4 = [1 + z(1)] hf_1, \quad (3.10)$$

$$h_5 = [1 + z(2)] hf_2, \quad (3.11)$$

$$h_{6m} = [1 + z(3)] hf_3, \quad (3.12)$$

in which x is the moisture content and z is a parameter used to modify the heat content of liquid water, depending upon the efficiency of the flash tank. Upon solving the above equations, we have

$$m_1 = \frac{y(1) h_o - [1 + z(1)] hf_1}{[1 - x(1)] hfg_1 - z(1) hf_1} \quad (3.13)$$

$$\frac{m_2}{m_4} = \frac{y(2) [1 + z(1)] hf_1 - [1 + z(2)] hf_2}{[1 - x(2)] hfg_2 - z(2) hf_2} \quad (3.14)$$

$$\frac{m_3}{m_5} = \frac{y(3) [1 + z(2)] hf_2 - [1 + z(3)] hf_3}{[1 - x(3)] hfg_3 - z(3) hf_3} \quad (3.15)$$

where

$$y(1) = \frac{h_o - q_1}{h_o} \quad (3.16)$$

$$y(2) = \frac{[1 + z(1)] hf_1 - q_2}{[1 + z(1)] hf_1} \quad (3.17)$$

$$y(3) = \frac{[1 + z(2)] hf_2 - q_3}{[1 + z(2)] hf_2} \quad (3.18)$$

The total work done per pound of well water is the sum of the products of turbine efficiency η , mass flow rate m and isentropic work Δh_s ,

$$\text{Total work} = \sum \eta m (\Delta h_s). \quad (3.19)$$

The values of the enthalpies of saturated water are functions of temperature. In the range of 150°F to 380°F, the following two interpolation equations were found by least square method:

$$h_f, \text{ Btu/lb} = -430.622108 + 0.8109447839T + 0.146245395 \times 10^{-3}T^2, \quad (3.20)$$

$$h_{fg}, \text{ Btu/lb} = 1183.89061 + 0.529042078 \times 10^{-2}T - 0.4812738654 \times 10^{-2}T^2 \quad (3.21)$$

where T is the absolute temperature in degrees Rankine. The corresponding standard errors of estimate are 0.095818758 and 1.3655710 Btu/lb respectively. The isentropic work can also be expressed in terms of flashing temperatures if the condensing temperatures of turbine and the moisture content of the flashed vapor are given; for example, at 120°F condensing temperature, the following two empirical equations were determined as functions of absolute temperature for flashed vapors at saturated condition and at 3% moisture content respectively:

$$\begin{aligned} \Delta h_s, \text{ Btu/lb} = & -2171.27186 + 6.3073419T - 0.05344237355T^2 \\ & + 0.1602543655 \times 10^{-5}T^3, \end{aligned} \quad (3.22)$$

$$\begin{aligned} \Delta h_s, \text{ Btu/lb} = & -1657.42887 + 4.211654994T - 0.2512764913 \\ & \times 10^{-2}T^2 + 0.3221495851 \times 10^{-6}T^3. \end{aligned} \quad (3.23)$$

The respective standard errors of estimate are 1.4406118 Btu/lb and 1.539438 Btu/lb. With the h_f , h_{fg} and Δh_s represented as functions of temperature, the total work became a function of three independent variables T_1 , T_2 , and T_3 . The process of determining the optimum flashing temperatures involves cumbersome calculations. The steepest ascent method was applied to the calculations with the aid of a digital computer.

For saturated well water at 400°F and condensing temperature of 120°F, the total turbine work per pound of well water is tabulated in Table 1 for one to 4-stage plants. With reasonable amounts of heat losses through the flashing tanks, Hansen's rule of approximation yields nearly the same amount of work as obtained by the exact method. As expected, the power contribution of an additional stage diminishes as the number of stages increases. The extra work outputs due to second, third and fourth stages are

Table 1. Flashing Temperatures and Total Work for Saturated Water
at 400°F and Condensing Temperature at 120°F.

Index	t_1 °F	t_2 °F	t_3 °F	t_4 °F	Work, Btu Per Pound of Well Water
1	280	-	-	-	22.377
	260	-	-	-	23.085°
	240	-	-	-	22.89
	220	-	-	-	21.743
2	307	213	-	-	29.33°
3	330	260	190	-	32.034°
	326.6	257.1	186.04	-	32.068
	322.91	255.23	183.31	-	32.084*
	313.94	248.04	173.94	-	31.979
4	344	288	232	176	33.173°
	335.24	276.37	216.02	170.93	33.681
	332.56	275.0	218.71	167.80	33.713*
	334	168.65	221.46	161.67	33.615
5	330	260	190	-	31.193°
	322.02	253.57	185.33	-	31.267
	316.83	247.10	180.75	-	31.271*
	304.92	224.22	172.18	-	30.962
6	330	260	190	-	27.035°
	315.12	247.25	181.36	-	27.513
	300	228	169	-	27.7 *
	287.6	173.8	132.64	-	20.563
7	330	260	190	-	17.088°
	301.67	253.3	174.09	-	19.571
	267	192.29	146.39	-	20.892*
	254.62	173.8	132.64	-	20.562

° Equal distribution of temperature difference.

* Optimal flashing temperatures.

1 1-stage flashing, no heat loss, zero moisture content of
flashing vapor, saturated water leaving tank.

Table 1. (continued) Flashing Temperatures and Total Work for
Saturated Water at 400°F and Condensing
Temperature at 120°F.

- 2 2-stage flashing, no heat loss, zero moisture content of flashing vapor, saturated water leaving tank.
- 3 3-stage flashing, no heat loss, zero moisture content of flashing vapor, saturated water leaving tank.
- 4 4-stage flashing, no heat loss, zero moisture content of flashing vapor, saturated water leaving tank.
- 5 3-stage flashing, no heat loss, 3% moisture content of flashing vapor, superheated water leaving tank ($z = 0.015$).
- 6 3-stage flashing, with heat loss ($y = 0.97$), 3% moisture content of flashing vapor, superheated water leaving tank ($z = .015$).
- 7 3-stage flashing, with heat loss ($y = 0.90$), 3% moisture content of flashing vapor, superheated water leaving tank ($z = 0.015$).

Turbine efficiency is assumed to be 75%.

29.33, 9.39 and 5.01 percents of the work of single stage plant respectively. Also, the specific volume of steam increases rapidly to cause difficulty in turbine design. Thus the number of flashing stages is likely to be limited to two only. Because of the pressure drops through pipes and high cost of low pressure equipment, the most economic flashing temperatures should be slightly higher than the calculated optimum temperatures.

CHAPTER IV.

WELLHEAD PRESSURE

The wellhead pressure affects the sizes of pipeline and plant components, the total flow rate of fluid from well, and the specific steam consumption for power production. Table 2 gives some typical values of wellhead pressures at various geothermal plants. The range of pressures is 60 to 210 psig. Contini [13] stated that the wellhead pressure should be 50 percent of the closed-in pressure; however, this rule is not followed usually. James [28] showed that no such relationship exists. Bruce [11] proposed that for vapor dominated fields, if the shut-off pressure is around 450 - 480 psig, the wellhead pressure should be about 120 psig. He also showed that for a shut-off pressure of 180 psig, the turbine inlet pressure should be 65 psig. As to vapor flashing plants using two cyclone separators, he suggested the inlet pressures of a mixed pressure turbine to be 75 and 2 psig respectively. For a plant using primary steam alone, Einarsson [17] concluded that the production of power increases with decreasing wellhead pressure and that the lower limit of the wellhead pressure depends upon the sizes and costs of the wellhead equipment, steam pipe and turbine. The upper limit of turbine inlet pressure is set by the requirement that the wetness in the turbine exhaust should not be greater than 14 percent unless a special moisture removal device is provided.

The advantages of applying high wellhead pressures are: 1. low specific steam consumption; 2. small borehold diameter; 3. small diameter of the main pipeline; 4. small size of the turbine for the same power output; 5. low rate of scale formation in the well. Geothermal wells are subjected to scaling mainly due to the precipitation of silica compound and calcium carbonate. The rate of precipitation is related to pressure, temperature, rate of flow, chemical composition of the solution, and gas content in the vapor. One of the methods of reducing the precipitation in wells is to operate the well at an adequately high pressure at the expense of lowering the flow rate and power output.

Wellhead pressures most commonly used now are in the range of 70 to 100 psig. In some plants at Geysers, the wellhead pressures have been lowered from 100 to 70 psig to improve plant efficiencies. At Wairakei, a

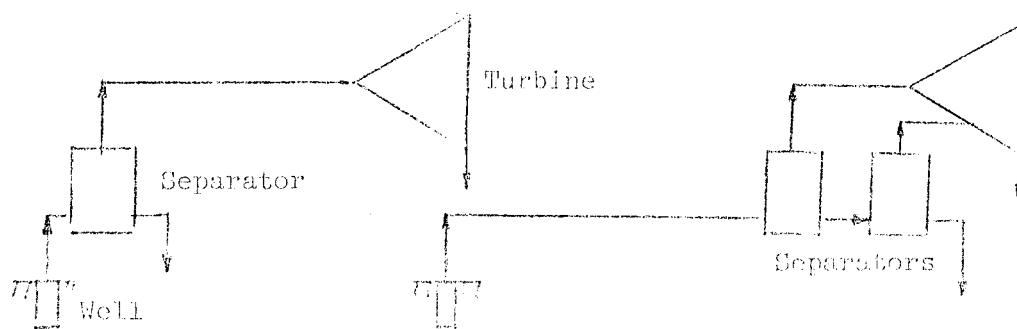
Table 2. Wellhead Pressures of Geothermal Plants.

	<u>Field</u>	<u>Location</u>	<u>Wellhead Pressure psig</u>
1	Vapor Dominated	Larderello, Italy	60
2	" "	Matsuda, Japan	65
3	" "	Geysers, U.S.A.	110 - 70
4	Liquid Dominated	Kamchatka, U.S.S.R.	37
5	" "	Otake, Japan	70
6	" "	Reykjavik, Iceland	75
7	" "	Wairakei, New Zealand	80, 140, 170, 210
8	" "	Cerro Prieto, Mexico	120

number of high pressure wells which originally discharged at pressures of about 220 psig are operating at 80 or 170 psig.

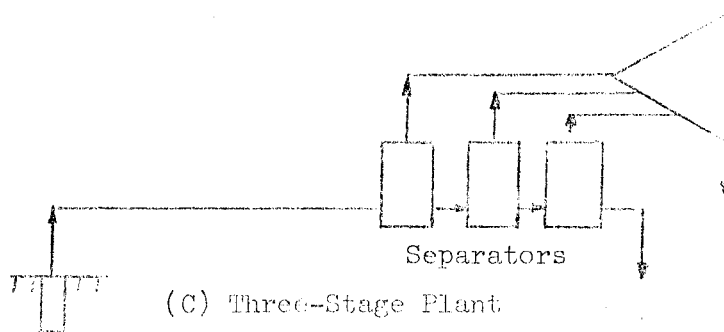
James [28] studied the problem of wellhead pressures by assuming three different cases. The first case is for vapor flashing plants using a single cyclone separator. For typical boreholes with the enthalpies of liquid-vapor mixture from 420 to 1200 Btu/lb, the optimum wellhead pressure varies from 70 to 100 psig if the pressure drop in the pipeline from the well to the plant is 25 psi. The second case is for steam filled aquifers where steam exists within a finite volume without an associated water phase, and the porosity of the volcanic rock varies from 0.1 to 1.0 [29]. The optimum wellhead pressure is calculated to be 70 psig, and it does not depend upon the initial closed-in pressure or the amount of volcanic rock associated with the steam. The third case is for a hot water aquifer, for which two possibilities exist. The steam may be drawn off from the top, or the pressurized hot water may be tapped from the bottom. The pressures that give the maximum operational life of the reservoir are 50 psig for the extraction of steam and 70 psig for the removal of hot water. Next he considered capital cost factor as a function of the wellhead pressure. The minimum cost occurs at the operating pressure in the range of 80 to 100 psig.

The power output can be enhanced by employing a multi-stage flashing plant. To evaluate the effect of wellhead pressure and number of stages on power output, the optimum wellhead pressures have been calculated in this study for different arrangements as shown in Figure 3. In the arrangement A, the steam is separated in a cyclone separator at the wellhead and the remaining liquid is rejected. In the other arrangements, the liquid vapor mixture is transported in a single pipe to the plant, where the steam is separated and flashed out in two or more stages. The typical flow rates of wells for the purpose of calculations are given in Figure 4, which shows that the well flow rate decreases as the wellhead pressure increases. The power output is the product of the flow rate and the work per pound of fluid, whose available energy decreases with the decrease of wellhead pressure. There exists an optimum wellhead pressure for the maximum power output of each arrangement. The optimum flashing pressures can be determined by using the rule discussed in Chapter III. The enthalpy of the mixture is assumed to be

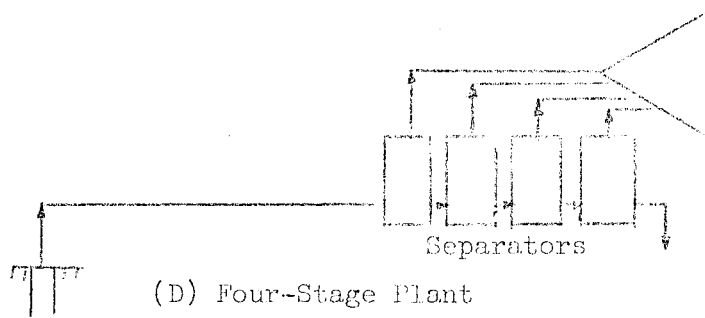


(A) Single-Stage Plant

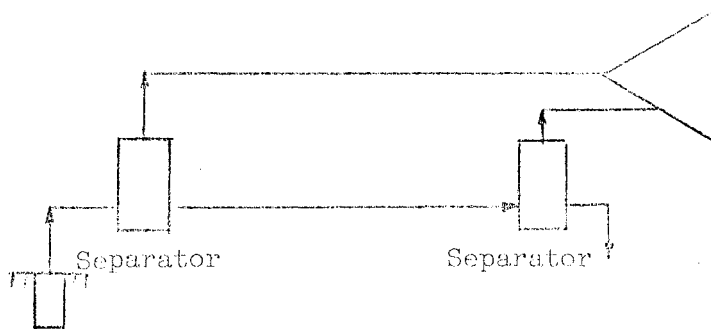
(B) Two-Stage Plant



(C) Three-Stage Plant



(D) Four-Stage Plant



(E) Two-Stage Plant with Separate Pipelines for Steam and Water

Figure 3. Arrangements of Vapor Flashing Plants.

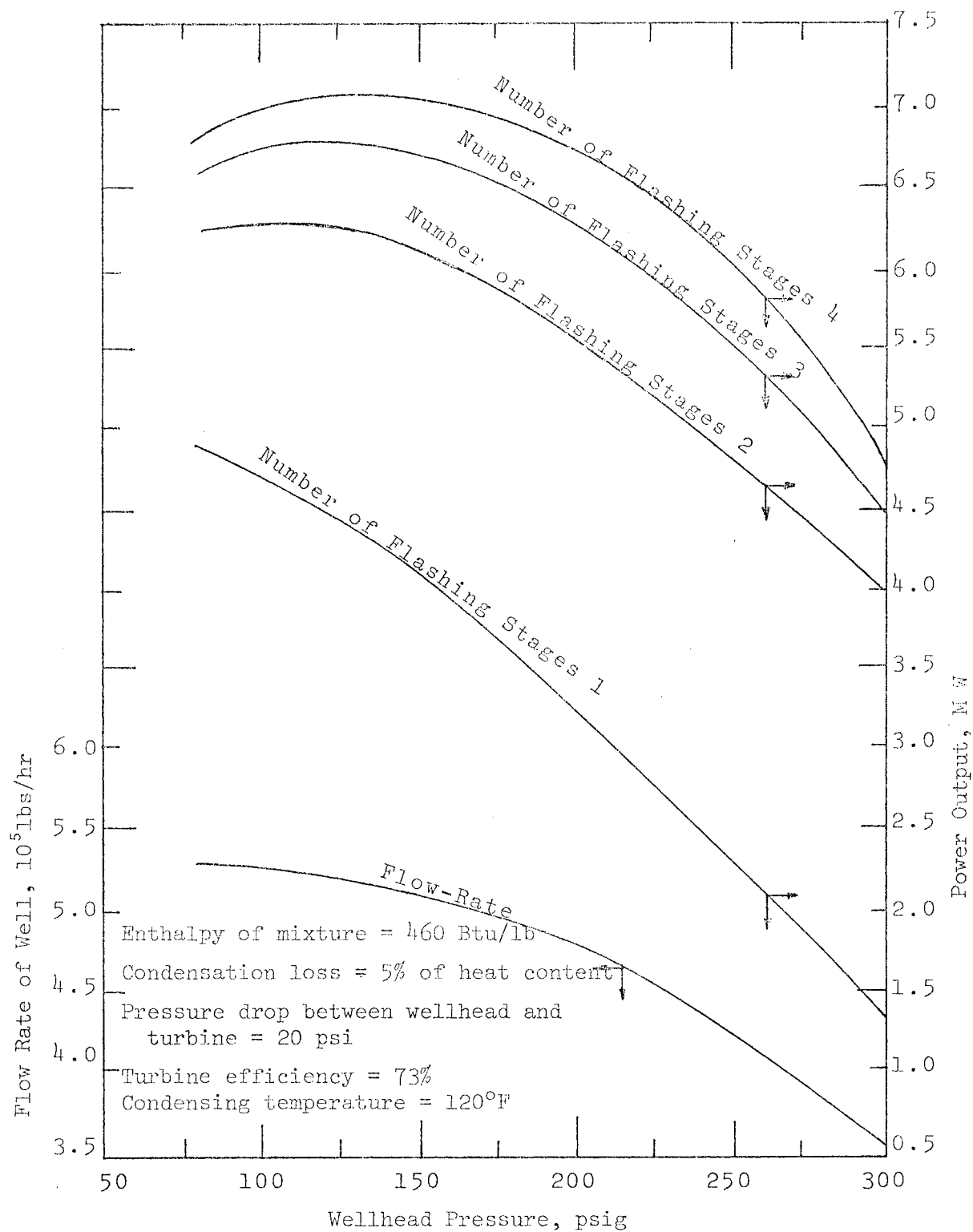


Figure 4. Well Flow Rate and Power Output Versus Wellhead Pressure.

460 Btu/lb. To account for the pressure drop and the condensation loss in the main pipeline, a pressure drop of 20 psi and a condensation loss of 5 percent of the fluid heat content are assumed. A combined turbine-generator efficiency of 73 percent has been used in this example. Under these assumptions, the variations of the gross power output for the four arrangements shown in Figure 3 are determined, and the results are given in Figure 4 in terms of wellhead pressure. The maximum power output of the arrangement A shown in Figure 3 is at the wellhead pressure of 80 psig, which is also the optimum pressure suggested by James [28]. By adding a flashing stage to the basic arrangement, the gain in power output is about 28.2% as shown in Figure 4. As the number of stages increases, the optimum wellhead pressure also increases and the rate of power gain diminishes. The second and third additional stages contribute relatively small gain in power output, 8.2 and 3.9% of the power output of the single stage plant respectively. The specific volume of the flashed steam and the size and cost of turbine increase rapidly with the number of stages. However, the moisture of exhaust steam in a multi-stage arrangement is lower than that in the single stage arrangement. The moisture contents and the specific volumes of the steam in the last flashing stages of the four arrangements of the example are as follows:

	Single Stage	2-Stage	3-Stage	4-Stage
v of steam ft ³ /lb	4.6	22.2	36.4	53.5
x, $\frac{\text{lb. water}}{\text{lb. mixture}}$	18.5%	16.5%	16.1%	15.8%

In consideration of the overall costs, the optimal number of stages might be two for the majority of the plants.

In the arrangement E shown in Figure 3, steam and water are separated at the wellhead and transported separately to the plant, where water is flashed to steam. Takahashi [42] and James [30] carried out their studies independently and concluded that the arrangement E is not as economical as the arrangement B, which can yield the additional power at the least cost.

Generally, a turbine is driven by steams from several wells. Each well has its own flow characteristic. The characteristic curves of the flow rate versus wellhead pressure for the individual wells may be superimposed to

obtain the combined characteristic curve in the same manner as dealing with the overall performance of pumps in parallel. The pressure drops in the branches and the main line are directly related to the power loss, the design and the cost of the piping system. In some existing systems, the pressure drop amounts to about 15-20 psi. When selecting the wellhead pressure, the effect of pressure drop on the system performance must not be neglected.

In Figure 4, the curves near the points of maximum power output are flat. It appears desirable to select a wellhead pressure higher than the optimum pressure since the sizes of pipes and equipment can be lowered by increasing the working pressure. However, Usui et al. [43] argued that it is not advisable to select a wellhead pressure higher than the pressure which gives the maximum power for the reason that such a design is apt to suffer power decrease by scale deposition on the nozzles and blades of the turbine. Another reason is that the well flow rate does not remain constant with time. At low wellhead pressure, the drop of flow rate may be retarded, and the underground paths of water are less likely to be blocked.

CHAPTER V.

CYCLONE SEPARATOR

A. Design and Performance

The main function of a cyclone separator in a geothermal plant is to separate the steam fraction from the steam-water mixture. A tangential inlet of the separator creates the spinning vortices in the fluid. The liquid water is centrifuged to the walls while the steam fraction is removed from a concentric outlet.

In the bottom outlet cyclone (BOC) separator shown in Figure 5, steam is removed from the bottom. At present, the BOC type of separator is very popular in vapor flashing plants. It can produce steam at a quality of 99.9 percent or better. A spiral inlet is better than a tangential inlet (Figure 5) because it can streamline the flow so that the erosion of the wall can be reduced. Bengma [8] conducted tests on a BOC separator to evaluate the performance and determine the optimum dimensions. He analyzed his results

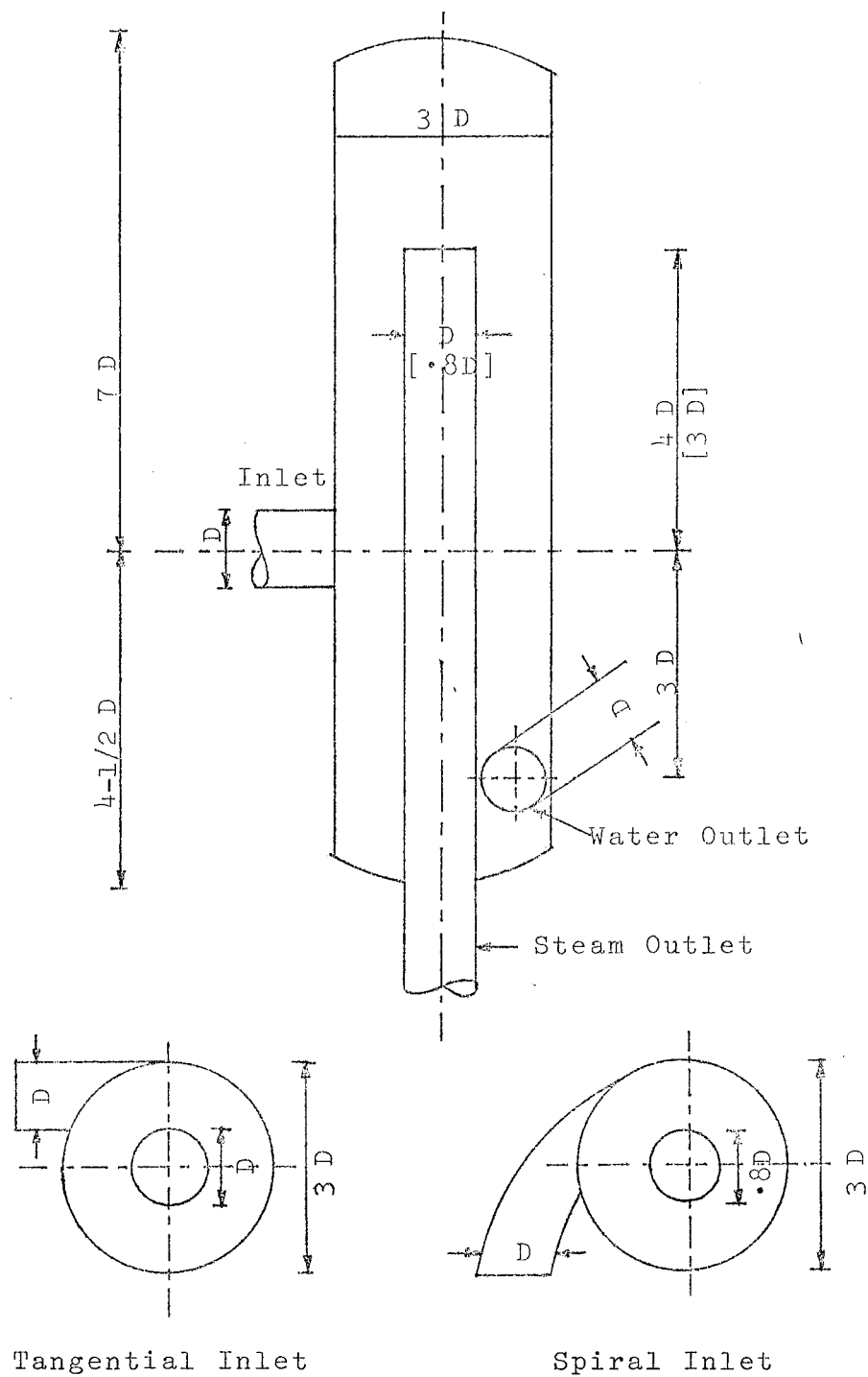


Figure 5 Bottom Outlet Cyclone Separator with Tangential and Spiral Inlets [8]

with inlet mass wetness, outlet mass wetness, inlet volumetric wetness, inlet velocity and breakdown point as the important parameters. The inlet and outlet mass wetness were defined as the ratio of mass flow of liquid water to the total mass flow of liquid water and steam. The inlet velocity was taken as the inlet mass flow of steam fraction multiplied by the specific volume of steam and then divided by the area of inlet. The breakdown point refers to the condition at which the mass wetness of separated steam is 0.5 percent. The following conclusions can be drawn from his investigation: (1) Separation is quite satisfactory at low inlet velocities regardless of the inlet wetness of steam. (2) The lower the separator pressure, the lower the inlet velocity at which breakdown occurs. (3) For a given separator inlet pressure, the drier the mixture, the greater the flow the separator can handle. (4) The pressure drop increases as the inlet velocity or the inlet volumetric wetness increases. Figure 6 shows the pressure drop between separator inlet and steam outlet plotted against the inlet velocity for different volumetric wetnesses. The variations of wetness of outlet steam with inlet velocity for different volumetric wetnesses are plotted in Figure 7.

To determine the most efficient separator configuration, the tests were made by Bengma [8] to experimentally find the optimum dimensions of the cyclone body, inlet, water outlet and steam outlet. In Figure 5 are the recommended dimensions, with the dimensions for tangential inlet in brackets. The dimensions are expressed in terms of the inlet diameter D since the inlet velocity governs the separator performance. From the rate of mass flow, the inlet mass wetness and the flashing pressure, the steam inlet velocity can be calculated for given values of pressure drop and separating efficiency by using Figures 6 and 7. The inlet pipe diameter and hence other dimensions can be determined from the steam inlet velocity and the recommended proportions in Figure 5.

B. Cyclone Controls

Figure 8 shows a possible arrangement of two cyclone separators. The liquid-vapor mixture from the wells are led to the main pipeline. To meet varying load demands in a narrow range, a pressure control valve is provided to regulate the flow rate. The operating pressures of the separators are controlled by two throttling orifices. The experiences of plant operation in

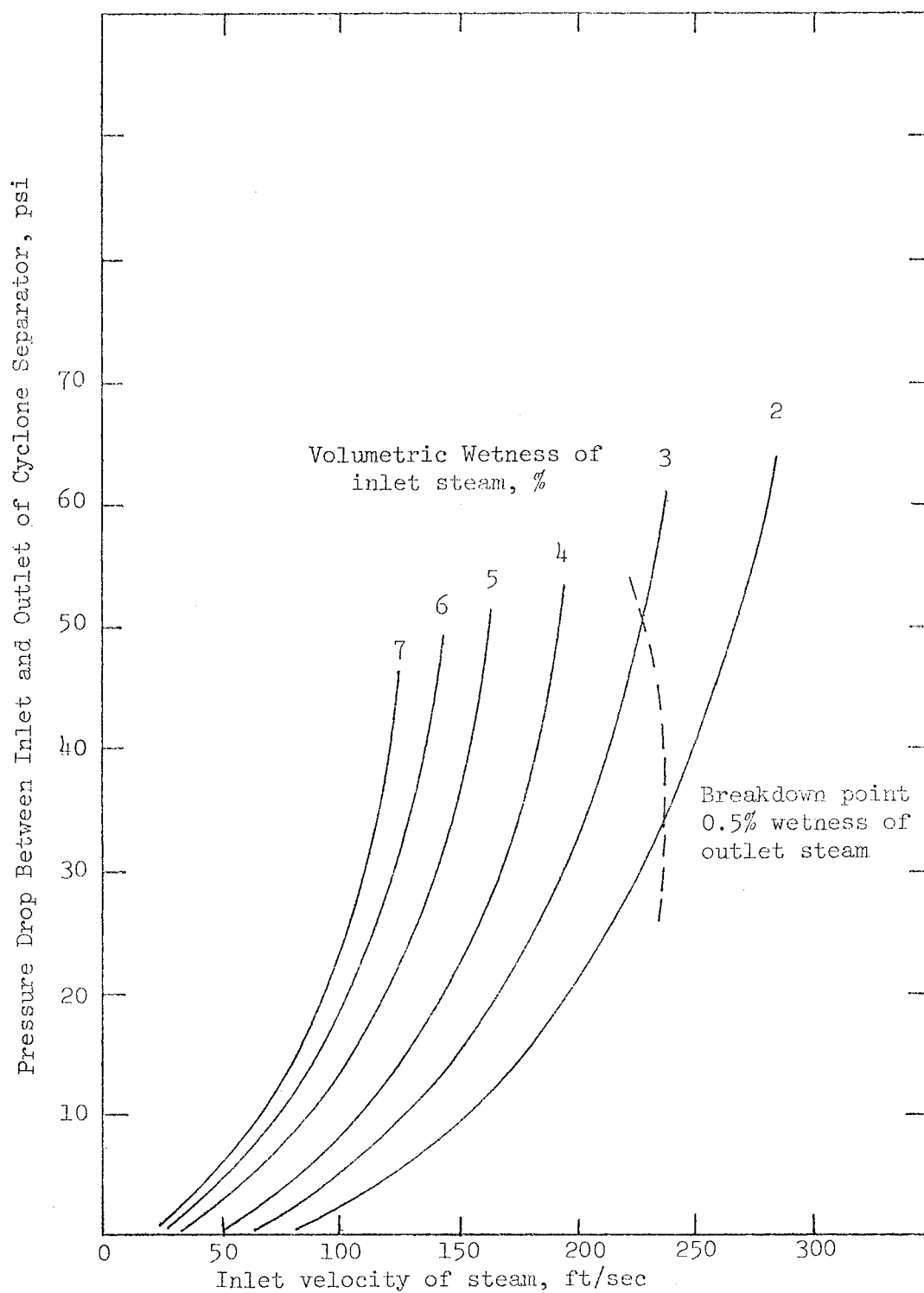


Figure 6. Effect of Inlet Wetness and Inlet Velocity on Pressure Drop Between Inlet and Outlet of BOC Separator [8]

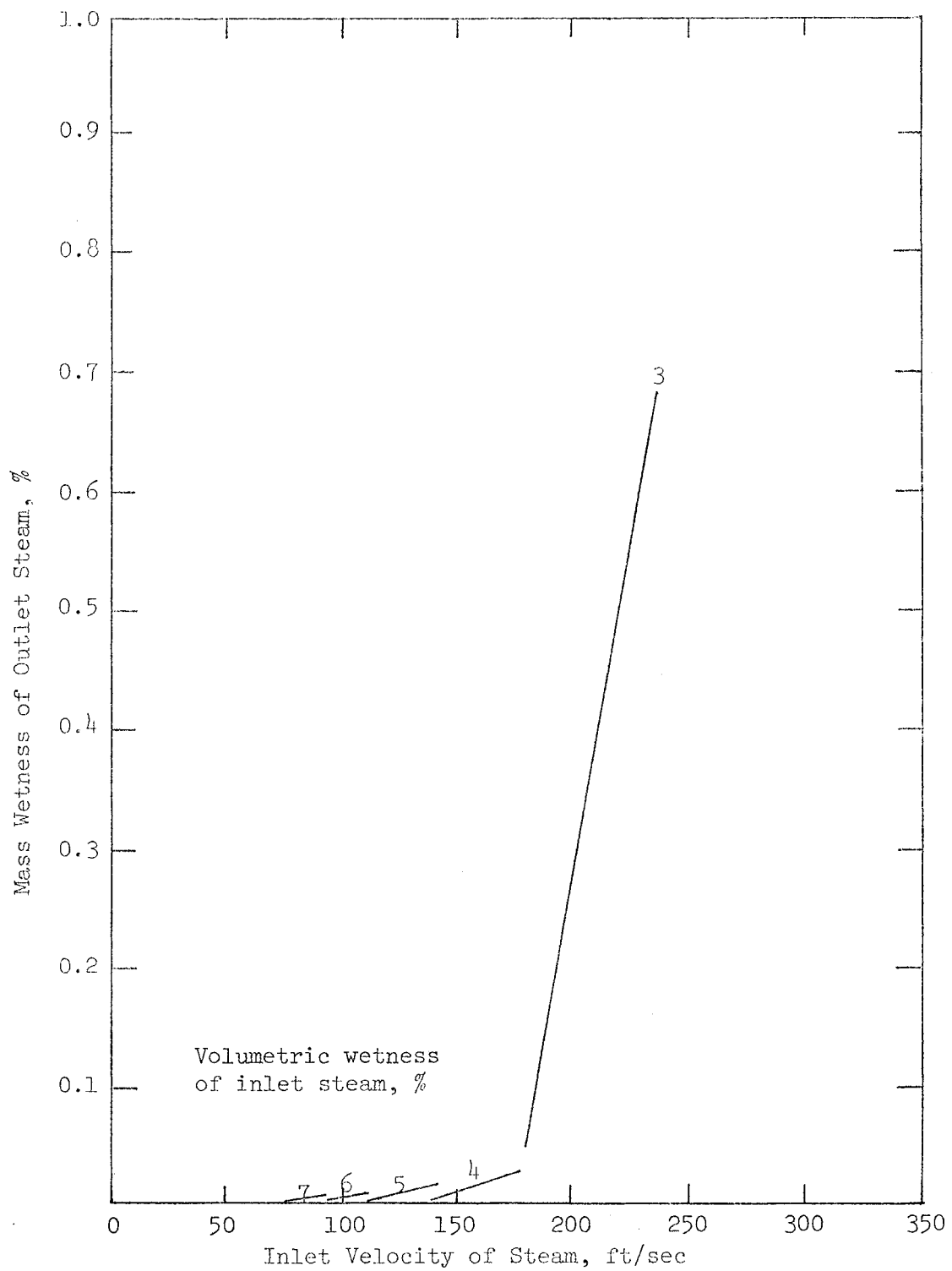


Figure 7. Effect of Inlet Wetness and Inlet Velocities on Wetness of Outlet Steam [8]

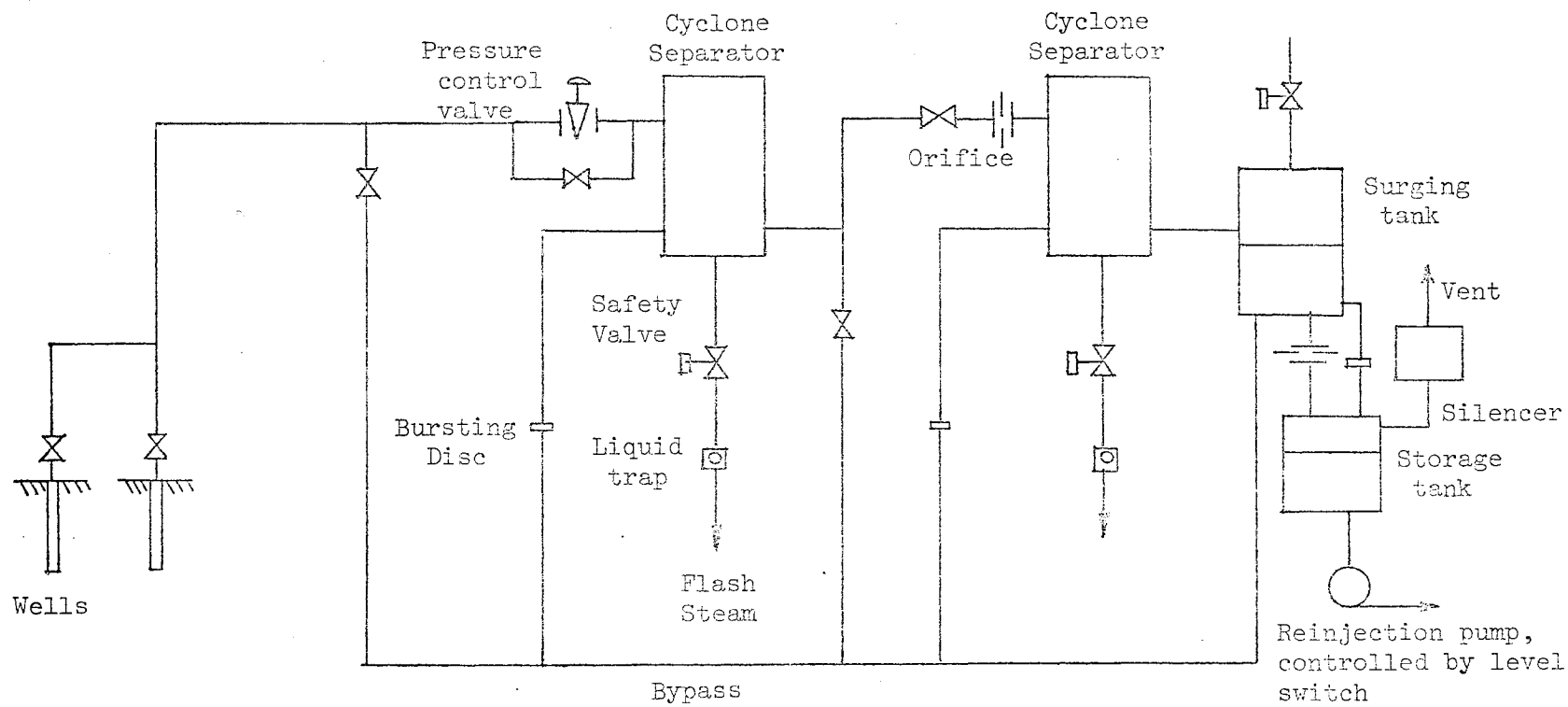


Figure 8. Flow Diagram of Two Liquid-Vapor Separators

New Zealand, Japan and Mexico show that a float controlled valve is not as reliable as a throttling orifice. If the pressure in the second separator is below atmospheric pressure, a steam ejector or vacuum pump should be used for maintaining the vacuum in the storage tank. The liquid trap in the diagram is a device which is used to check the flow of liquid in the steam line. Care must be taken to guard against any possibility of a slug of water entering steam line [5, 20, 44]. The water is collected in a surging tank, in which the level of liquid is effectively maintained by the throttling orifice. In the case of blockage of orifice or a large increase in water flow, the safety devices come into action for protection against the overflow in the surging tank.

CHAPTER VI.

TURBINE

In New Zealand, cross-compound turbine-generators have been used. At the three pressure levels, there are three sets of turbine-generator mounted on different shafts: high pressure topping set, intermediate pressure set and low pressure condensing set. The interconnecting pipes are incorporated with centrifugal separators which restrict the moisture content of the steam entering a turbine to one percent. To protect against excessive back pressures, each back pressure turbine is fitted with a back pressure limiting device. Lately, multiple inlet mixed-pressure turbines were installed to use the flash steams at the same pressure levels. For vapor flashing plants, it is more economical to have a mixed-pressure turbine.

Estimates of turbine-performance are usually required in the preliminary stages of power plant design, and the estimation can often be made from the information available in manufacturer's publications and other literature. As an illustration, the performance curves have been estimated for the 10 MW turbine in a three-stage vapor flashing plant which generates power from saturated hot water at 400 F. The temperatures of the flash tanks are 330, 265 and 200°F and the turbine back pressure is 4 inches Hg absolute. According to the technical bulletins by General Electric Company [19, 35], the ratio of

specific steam consumption at 20% to that at full load is 1.24. In Figure 9 is the estimated performance curve, and the basic steam rates can be found by extrapolation from the tables.

Turbine blading of 13 percent chromium steel has been used at Geysers to minimize erosion caused by the particulate matter in steam; however, Armstead [4] pointed out that this type of steel in the hardened stage is susceptible to stress-corrosion cracking. It is immune to stress-corrosion cracking in the soft non-martensitic state if sodium chloride and borates are absent or present in small quantities only. To remove carryover of salt, scrubbers are required to wash the flash steam after cyclone separators. The deposit of minerals on the internal surfaces of turbine can be very harmful because of two reasons. First, they impair the turbine efficiency and reduce the steam flow by closing the nozzle. Second, the heavy deposit can cause the failure of the shrouds of the blades due to excessive centrifugal force. The problem of deposits is especially serious in the dry high pressure region. The use of scrubbers and the adoption of soft chromium steel should alleviate these problems. Scrubbers may not be required if the steam velocity in the cyclone separator is less than 100 ft/sec [31].

As the saturated steam passes through the turbine, partial condensation takes place and the steam becomes increasingly wet as it approaches the exhaust end. The condensate which is mixed in the steam travels at high velocity in the form of small droplets. The impact of the droplets on metal surfaces can cause erosion, particularly in the moving blades of the last few stages of the turbine where the blade tip velocity is the highest. The generally accepted level of steam wetness is 13 to 14 percent by weight [45]. The amount of steam condensed depends upon the inlet and exhaust pressures. The greater the inlet pressure or the lower the back pressure, the higher is the steam wetness. The problem is not so acute with superheated steam since the condensation does not occur until the steam has given up its superheat. The soft blades are susceptible to erosion in the wet low-pressure end of the turbine. Moisture extraction buckets may be installed to remove some moisture from the steam. The buckets of the low-pressure stages may be coated with erosion resistant materials as an additional safeguard [1]. To avoid excessive erosion, the blade tip speed is limited to 900 ft/sec at Wairakei. A high ratio of blade

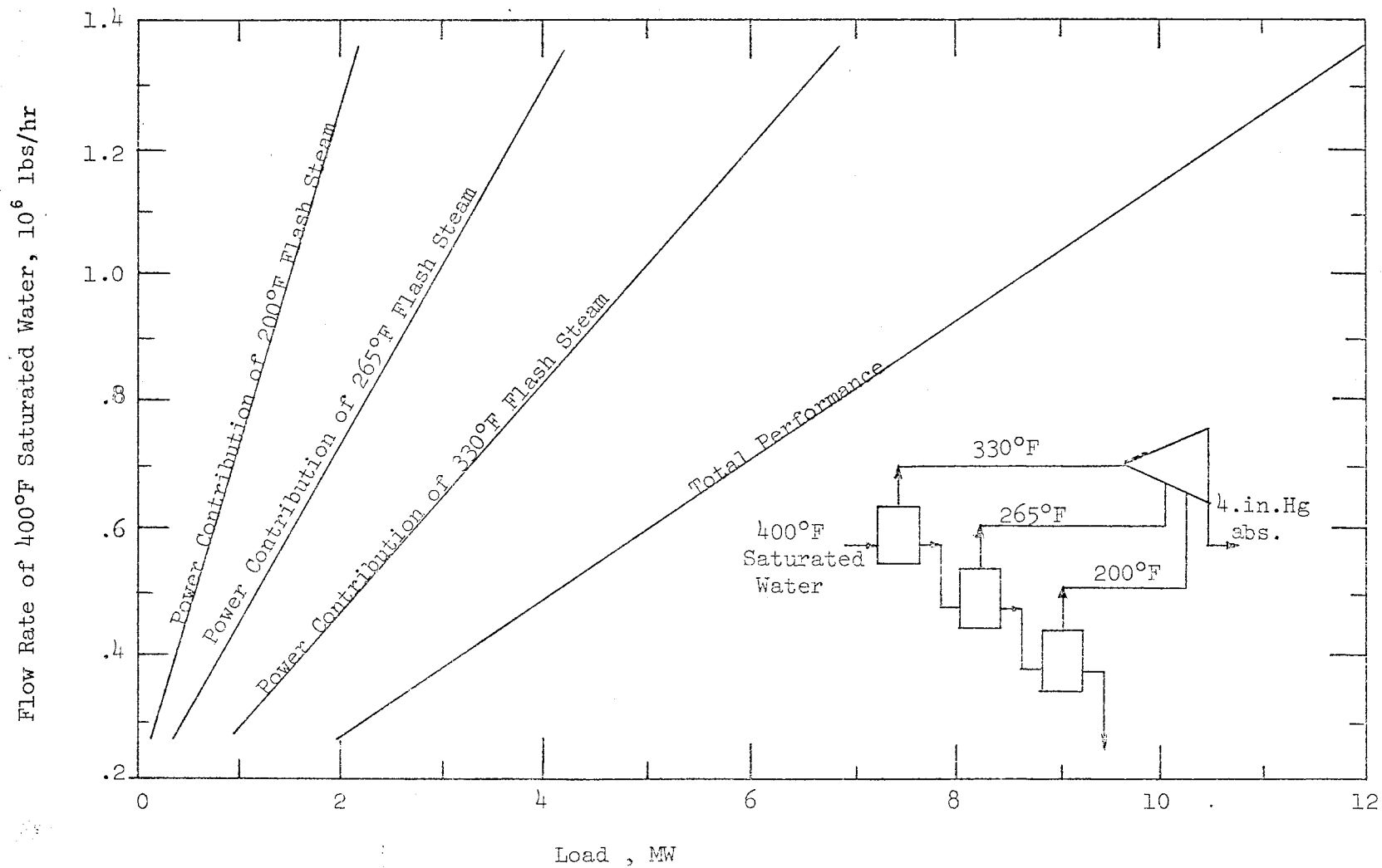


Figure 9. Estimated Performance of a 10 MW Turbine for Plant Using 400°F Saturated Water

velocity to steam velocity can help the separation of the condensate from the steam by centrifugal force. The condensate in the steam lines should be removed by liquid traps, which may be made of ball-float type of valve. Water and solid particle detectors may be installed in the steam lines to detect the presence of water or solids. In the case of emergency, the distress signal is given in the control room, and the turbine is tripped off.

Since the dust in the steam may cause the ordinary throttle valve to stick, it is recommended to use the butterfly valve for regulating the flow, as practiced at Geysers. A swing check valve may be used as the stop valve in the main line. A trip check valve may be installed between the stop valve and the regulating valve to provide additional overspeed protection. A y-type strainer should be installed before the main valves to catch foreign materials.

CHAPTER VII.

HEAT REJECTION EQUIPMENT

A. Types of Plants and Condensers

In a non-condensing plant shown in Figure 10, the steam emitted from the well is admitted directly to the turbine and is exhausted to the atmosphere. It is the simplest plant as it does not require a condenser and its auxiliary equipment. Both the capital investment and its operating expenses are low. The main drawback is its high steam consumption. For the same power output, the consumption of a non-condensing plant may double the amount of steam required by a condensing plant. As pointed out by Chierici [12], the installation of a non-condensing plant is advisable only when a high percentage of non-condensable gases is encountered in the exhaust steam.

When the gas-content is lower than 8 to 10 percent, a condensing plant is recommended by Chierici [12] and Zancani [47] as shown in Figure 11. To maintain a high vacuum in the condenser, the non-condensable gases must be removed with special gas-extractors. All the large plants in use are of the condensing type.

An experimental aluminum-tube surface condenser was installed in Italy [47] Owing to corrosive properties of the natural steam, it ended in a complete failure. The presence of H_2S and NH_3 in steam eliminates the possibility of using carbon

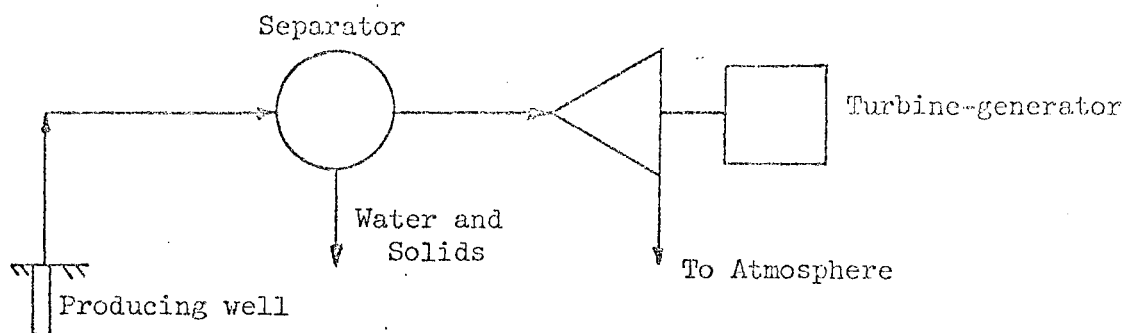


Figure 10. Non-Condensing Plant

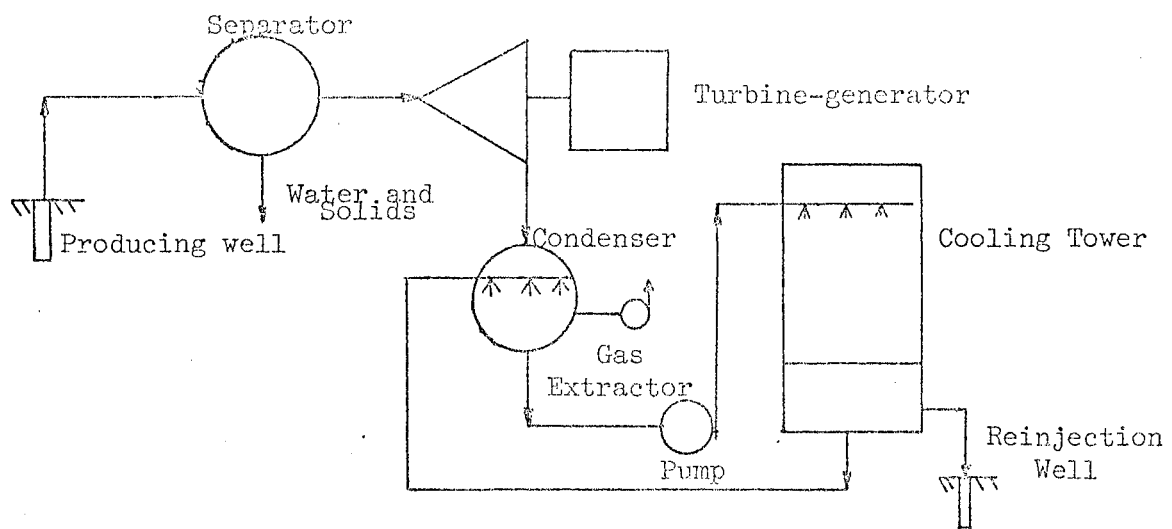


Figure 11. Condensing Plant

steel, copper, aluminum and nickel alloys. Only stainless steel is suitable. The non-condensable gases in surface condensers can lower the heat transfer coefficient and affect the required sizes of condenser. Since geothermal plants do not operate in a closed cycle, there is no need to recycle the condensate. Thus, a direct contact condenser could be more suitable than a surface condenser. In a direct contact condenser, the steam condenses by mixing with cooling water. Zancani [47] compared the cost and performance of the two types of condensers from the standpoints of power production and recovery of chemical substances for the basic condensing plant and the plant using secondary steam. His conclusion is that the direct contact condensers are better than the surface condensers. The major disadvantages of surface condensers are high initial cost, large size and poor heat transfer.

B. Barometric Condenser

There are two types of direct contact condensers, the barometric and low-level jet. In a barometric condenser (Figure 12), the steam enters at the bottom and rises with decreasing velocity while the cooling water falls down in fine droplets from the top. The cooling water and condensate discharge through a vertical tail-pipe by gravity. The tail-pipe is extended into the hot well, from which the condensate overflows. Lyle [32] suggested that the capacity of the hot well should be greater than the volume of the tail-pipe to prevent the seal from being broken. Heat Exchange Institute Standard [40] specifies the tail-pipe seal and clearance from tail-pipe outlet to the bottom of hotwell as shown in Figure 12. If the condenser and the cooling tower are properly located, the vacuum head alone in the condenser could cause water to flow from the cooling tower basin to the condenser.

The temperature of condensate t_2 cannot exceed that of the incoming vapor, t_v . In an ideal condenser, $t_2 = t_v$ and the required amount of cooling water for a given condition of steam is a minimum. The difference $t_v - t_2$ represents the degree of deviation from the maximum efficiency and is called the approach. In a counter-current condenser, the recommended approach for design purposes is given by Hugot [24]:

$$t_v - t_2 = (0.1 + 0.02a) (t_v - t_1) \quad (7.1)$$

where a is the percentage of non-condensable gases in the entering mixture of

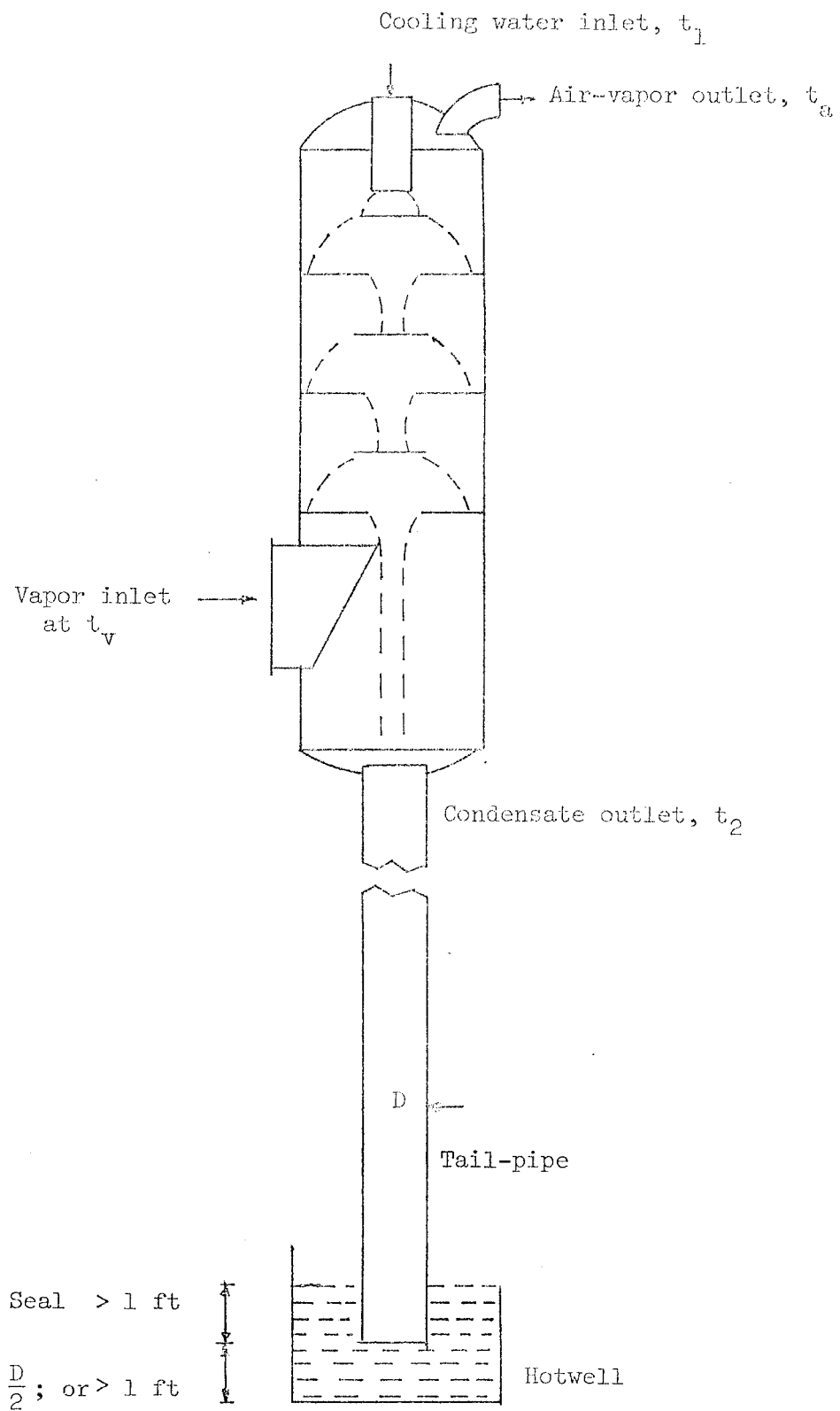


Figure 12. Counterflow Barometric Condenser

steam and gases by weight and t_1 is the temperature of entering cooling water. The minimum value of approach should be 5°F.

The temperature t_a of the leaving non-condensable gases lies between the temperature t_v of the gaseous mixture and the temperature t_1 of the cooling water. In a counter-flow condenser t_a is approximately equal to t_1 .

The curtain of cooling water offers resistance to the passage of vapor and incondensable gases. The resistance in a properly designed condenser should not exceed 0.15 inches of water. By neglecting the flow resistance, the pressure p_c in the condenser may be considered constant throughout the condenser. The value of p_c corresponds to the vapor pressure of water at the temperature t_v of the entering vapor and is equal to the sum of the partial pressure of water vapor p_v and the partial pressure p_a of the non-condensable gases at the various sections of the condenser. As the vapor condenses, its partial pressure decreases and the partial pressure of non-condensable gases increases. At the outlet, the non-condensable gases are saturated with water vapor, and the partial pressure of non-condensable gases is

$$p_{a1} = p_c - p_{v1} \quad (7.2)$$

where p_{v1} is the partial pressure of water vapor and is approximately equal to the vapor pressure of water at the cooling water temperature t_1 . The weight of water vapor W_{v1} in the gaseous mixture can be determined from equation:

$$W_{v1} = W_a \times \frac{MW_v}{MW_a} \times \frac{p_{v1}}{p_{a1}} \quad (7.3)$$

where W_a is the weight of non-condensable gases, and MW_v and MW_a are the molecular weights of vapor and non-condensable gases respectively. The volume of gaseous mixture V_m may be found by the relations,

$$V_m = \frac{W_a \times R_a \times T_1}{p_{a1} \times 144} = \frac{W_{v1} \times R_v \times T_1}{p_{v1} \times 144} \quad (7.4)$$

where R_a and R_v are the gas constants of non-condensable gases and vapor respectively.

C. Gas Extractors

To maintain high vacuum in a condenser, the non-condensable gases must be removed continuously from the condenser. There are four types of gas extractors which have been used in geothermal plants: steam jet ejector, water jet ejector, reciprocating pump, and centrifugal compressor. Steam jet ejectors and reciprocating pumps are suitable for plants which do not require the removal of large quantities of non-condensable gases, such as plants at Wairakei, Geysers, Matsuda and Otake, where the gas-contents in steam are less than one percent by weight. The steam ejectors are simple in construction and operation; however, they are not economical in energy consumption although their initial cost is very low. At Geysers, the ejector has two stages with the motive steam at the inlet pressure of turbine. The first-stage-ejector compresses the gases from the condensing pressure of 4 inches Hg. absolute to about 5 psia; the discharge from the second stage is at about 1.07 atmospheres. The inter and after condensers, used to condense steam in the gaseous mixture from the ejectors, are of the barometric type. The ejectors consume about 5 percent of the full-load steam flow to turbine. The only geothermal plant which uses a motor-driven reciprocating pump is at Otake, Japan. It is far more efficient in energy consumption than the steam ejector but has a high maintenance cost.

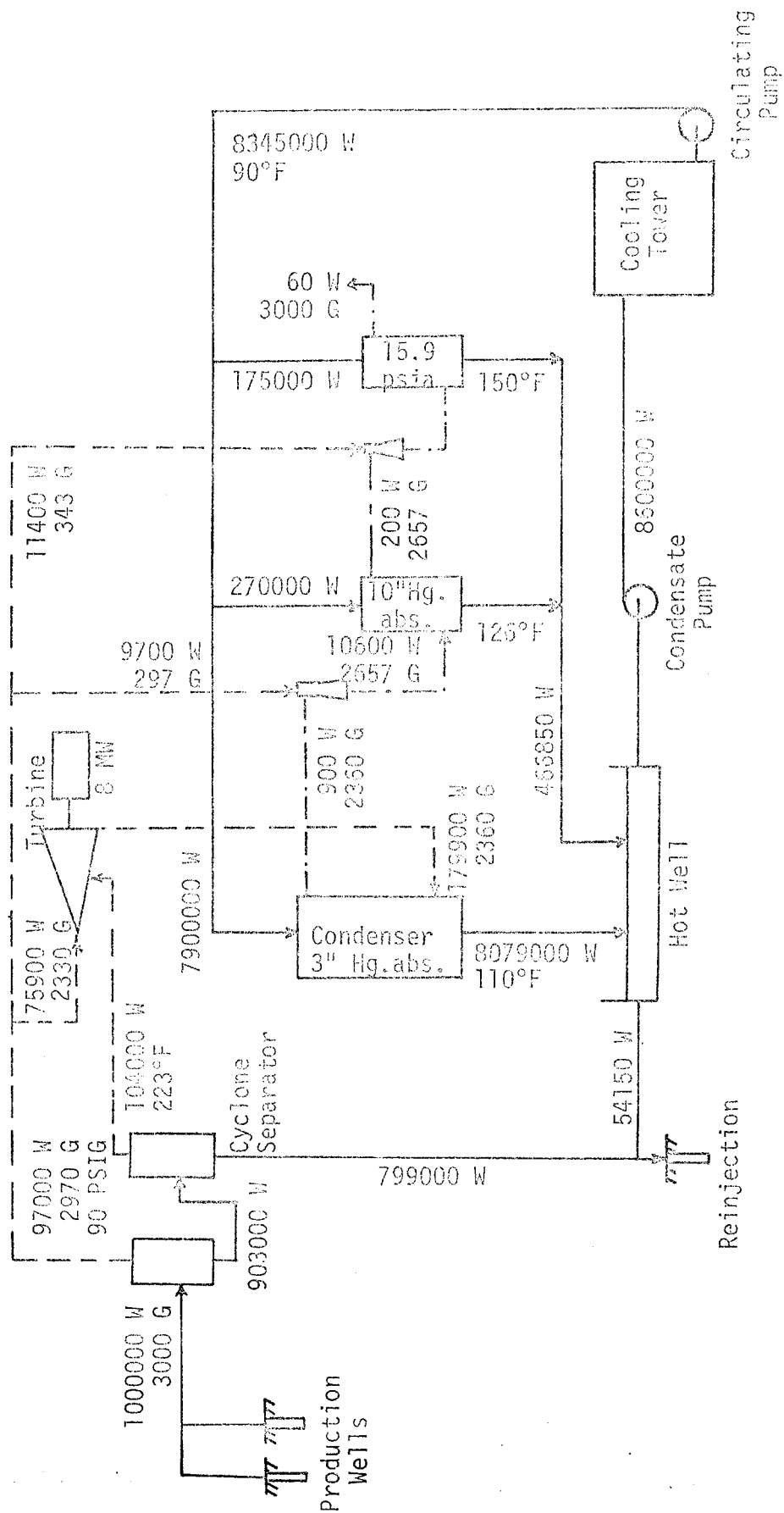
In the plants at Larderello, the centrifugal compressors are found to be effective in removing a large quantity of non-condensable gases at high concentration (about 4%). Di Mario [16] and Dal Secco [15] described the construction and operation of the compressors, which can compress 85,000 to 100,000 m³/hr of saturated gas at 30°C from .08 to 1.04 atm. The compressors are driven by 3 or 4 speed electric motors or auxiliary turbines which can be operated at variable speeds according to the operating conditions such as vacuum in the condenser, steam flow and gas-content. The compressor may also be geared to the shaft of the turbo-generator, operating at constant speed with adjustable blades. At part loads, a variable-speed drive consumes less power. To save mechanical work, counterflow barometric intercoolers are used to cool the gases. The carryover of condensate droplets in the gaseous mixture can cause the corrosion of structural materials of compressors and the fouling of impellers and diffusers. By placing baffles above the water sprayers in the intercooler and by properly dimensioning the coolers to limit the speed of flow, the carryovers are minimized.

Due to the simultaneous presence of hydrogen sulfide and moisture, the gas-extracting equipment is prone to corrosion. The intercoolers may be made of carbon-steel lined with a layer of lead as practiced at Larderello or covered with resinous coatings as done at Wairakei. At Geysers, the condensate and gas-pipes are made of 18-8 stainless steel, and the pumps are built with clad stainless steel (13 percent chrome, 14 percent nickel and 2-1/2 percent molybdenum).

D. Heat Balance

The design of a plant starts from the energy and mass balances, or called heat balances, from which the system performance and the requirements of the components may be determined. The first step of heat balance is to select basic parameters. By varying the values of the parameters, a number of heat balances can be worked out to form the basis for making decisions on the selection of the design. As an example, the heat balance of a two-stage vapor flashing plant is given in this section. Figure 13 shows the flow diagram of the plant, for which the basic assumptions are as follows:

Power output	=	8 MW
Number of cyclone separators	=	2
Wellhead pressure	=	100 psig
Number of wells	=	2
Water flow	=	10^6 lb/hr
Gas flow	=	3000 lb/hr
Enthalpy of water	=	400 Btu/lb
Pressure drop in main pipeline	=	10 psi
Condensation loss	=	3% of the heat content of water
Turbine-generator efficiency	=	73%
Pump efficiency	=	75%
Fan motor efficiency	=	90%
Ejector efficiency	=	5.6%
Condensing pressure	=	3 in. Hg. abs.
Wet bulb temperature	=	75°F
Approach of cooling tower	=	15°F



W = lb. water and steam per hr., G = lb. gas per hr.

Figure 13. Flow Diagram of a Vapor Flashing Plant

The process of heat balance involves the selection of the optimum flashing temperature, the energy and material balances of geothermal water, non-condensable gases and cooling water. The optimum flashing temperature of this example is 223°F, as determined by the rule stated in Chapter III. After the temperature in the second cyclone separator is selected, the steam productions of the two separators can be determined. They are 97,000 lb/hr and 104,000 lb/hr respectively.

To perform the gas balance, it is necessary to estimate the solubility of the non-condensable gases in hot water. Since nearly 95 percent of the non-condensable gases from a typical well is carbon dioxide, the properties of non-condensable gases may be approximately represented by those of carbon dioxide. In a dilute solution, Henry's law is applicable. It states that the mole fraction N_g of a gas dissolved in a liquid in equilibrium condition is directly proportional to the partial pressure P_g of the gas above the liquid surface,

$$P_g = K \cdot N_g \quad (7.5)$$

where K is the Henry's constant, a function of temperature. For water and carbon dioxide system, if the partial pressure is expressed in kg/cm^2 , then K may be expressed by the following equation in terms of temperature t in °C as proposed by Mallin [33]:

$$K = 39.66 + 67.7403 t - 0.1788t^2 \quad (7.6)$$

When the mixture of water and carbon dioxide is in phase equilibrium at a moderate pressure (Figure 14), the mole fraction N_g and the partial pressure P_g of carbon dioxide are governed by the relations,

$$N_g = \frac{M_{g1}/44}{M_{l1}/18 + M_{g1}/44} \approx \frac{18}{44} \frac{M_{g1}}{M_{l1}} \quad (7.7)$$

$$P_g = P \cdot \frac{M_g - M_{g1}}{44} \bigg/ \left(\frac{M_s}{18} + \frac{M_g - M_{g1}}{44} \right) \approx P \cdot \frac{M_g}{44} \bigg/ \left(\frac{M_s}{18} + \frac{M_g}{44} \right) \quad (7.8)$$

where M_l , M_s and M_g represent the mass quantities of liquid water, steam and gas respectively, and M_{g1} is the mass of gas dissolved in liquid. The above 4 equations were used for the gas balances given in Figure 13.

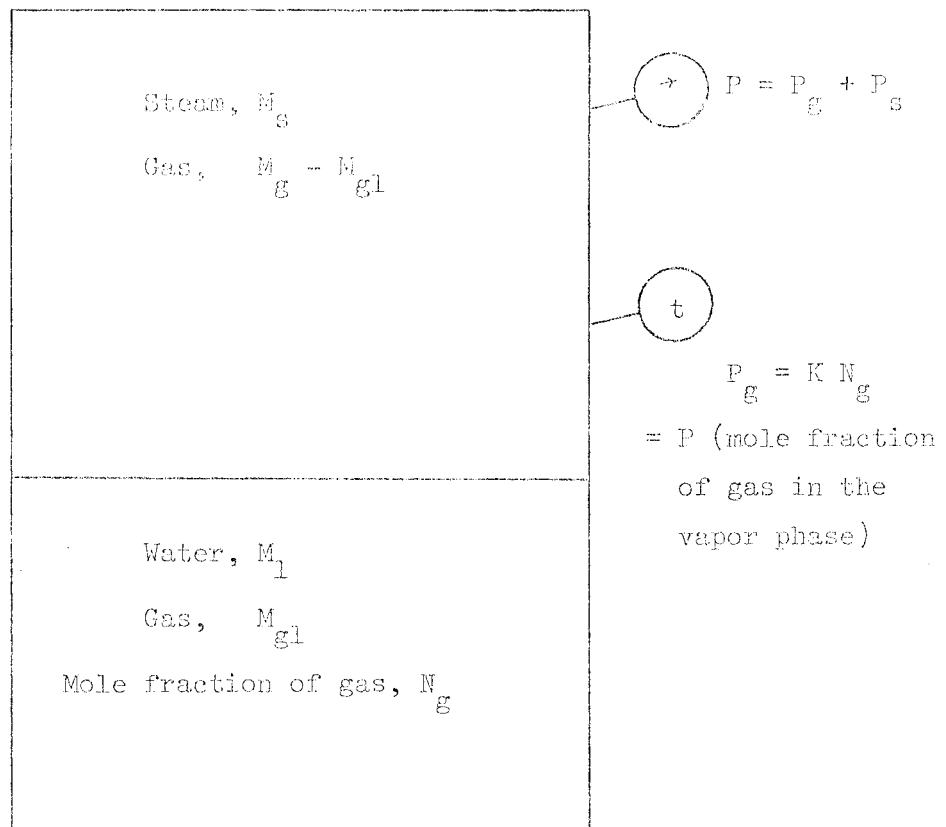


Figure 14. Gas and Water in Equilibrium

In the steam balances, it is necessary to evaluate the steam consumption e_1 and e_2 in pounds steam per pound gas removed by the first- and second-stage ejectors respectively. By referring to the symbols in Figure 15, the mass flow W_s of the high pressure steam can be related to e_1 and e_2 by the equation,

$$W_s = W_{1s} + e_1 (W_{1s} X_g + W_g) + e_2 [W_{1s} X_g + W_g + e_1 (W_{1s} X_g + W_g) X_g] \quad (7.9)$$

By rearranging the terms, one has

$$W_{1s} = \frac{W_s - W_g (e_1 + e_2) - e_1 e_2 W_g X_g}{1 + X_g (e_1 + e_2) + e_1 e_2 X_g^2} \quad (7.10)$$

The specific volume V_m of the gas-vapor mixture entering the ejector can be determined by equations (7.2) and (7.4). The isothermal work W per pound of gas for the compression of gas from the pressure p_1 to p_2 is

$$W = p_1 V_m \ln \frac{p_2}{p_1} \quad (7.11)$$

The ejector efficiency E is 5.6% in the given assumptions based on ejector consumption curve supplied by Dal Secco [15]. It is defined as the ratio of isothermal work per pound of gas W to the isentropic change of enthalpy Δh of the motive steam. By this definition, the steam consumption e of the ejector can be expressed as

$$e = \frac{W}{\Delta h \times E} \quad (7.12)$$

The values of e_1 and e_2 for the example were found to be 4.7 and 3.7 lb steam/lb gas respectively.

The approaches of main condenser, inter-condenser and after-condenser were determined from equation (7.1), and the heat transmission H_L of the main condenser is based on the relation,

$$\begin{aligned} H_L = & W_{1s} (ho_1 - hc) + W_{2s} (ho_2 - hc) + (W_{1s} + W_{2s} - W_v) (t_v - t_2) \\ & + (W_g + X_g W_{1s}) C_{pg} (t_v - t_1) - W_v (h_1 - hc) \end{aligned} \quad (7.13)$$

where W_v is the amount of vapor in the saturated gas-vapor mixture and can be determined using equation (7.3), ho_1 and ho_2 are the enthalpies of exhaust steams,

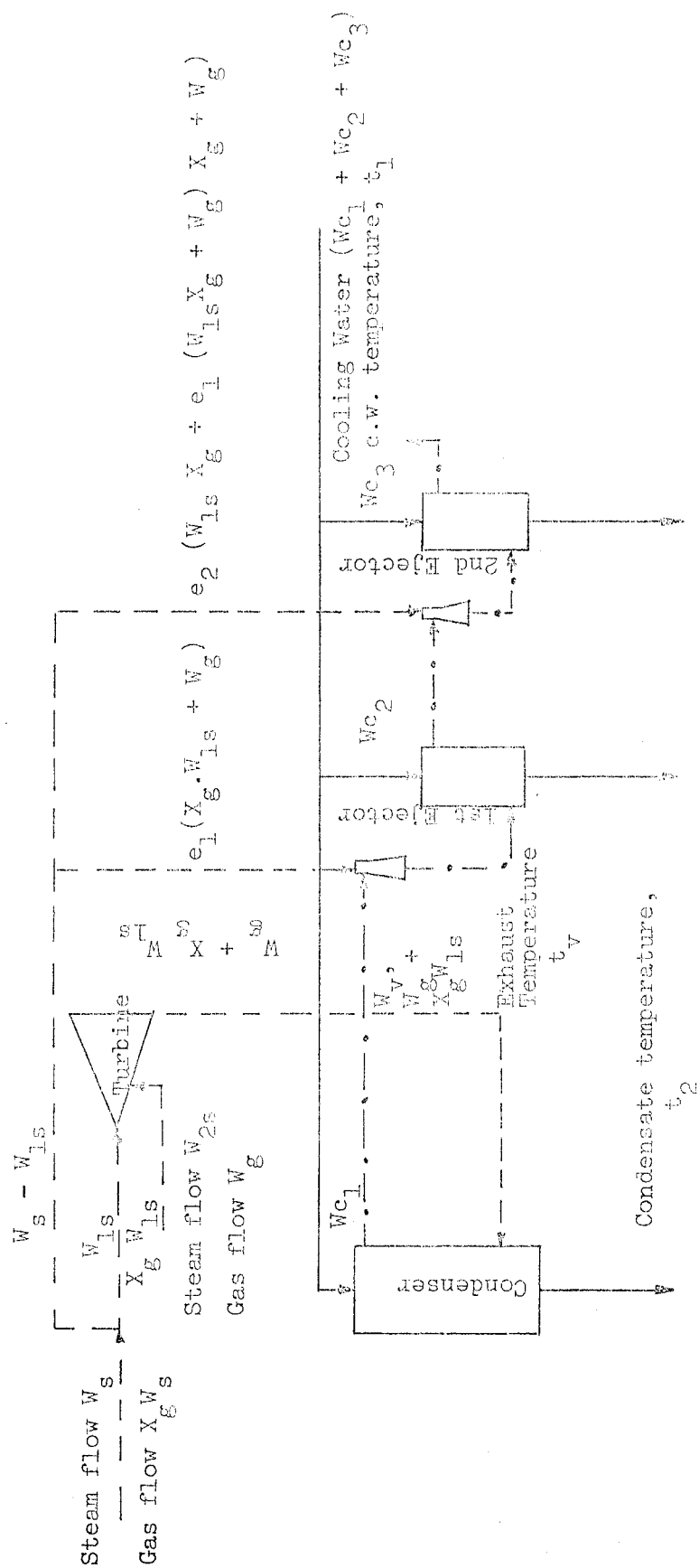


Figure 15. Steam Balance of Ejectors

h_g is the enthalpy of saturated vapor at temperature t_1 , h_c is the enthalpy of water at temperature t_v , C_{pg} is the specific heat of gas (0.202 Btu/lb) and the other symbols are given in Figure 15. The rate Wc_1 of cooling water to the main condenser is given by

$$Wc_1 = \frac{H_L}{(t_2 - t_1)} \quad (7.14)$$

Similarly the cooling water flow to inter- and after-condenser can be determined.

A cooling tower has two kinds of losses of water, evaporation loss and drift loss. According to Cootner et al. [14], the evaporative loss is about 1% of cooling water for each 10 degrees of cooling, and the drift loss is about 0.2% of water circulated through the tower when drift eliminators are used. The flow rate L of water to the cooling tower was determined by the equation,

$$L = (Wc_1 + Wc_2 + Wc_3) / [0.998 - 0.001 (t_{avg} - t_1)] \quad (7.15)$$

where t_{avg} is the average temperature of water in the hotwell. The fan horsepower for large cooling towers ranges from one to two times the value of 0.000014 KL as reported by Berg et al. [10], where K is the rating factor given in the American Society of Heating, Refrigerating and Air Conditioning Engineers Guide and Data Book [2]. The fan power was calculated by the equation $0.746 \text{ kw/hp} \times 0.00002 \text{ KL} \div 0.9 = 0.000016 \text{ KL}$, where the factor 0.9 represents the motor efficiency as specified in the basic assumptions. The power consumption of the fan was found to be 175 kw. The power required by the condensate pump is 250 kw if the pump head is assumed to be 60 ft, and the power absorbed by the circulating pump for a head of 32 ft is 135 kw. Out of the gross power output of 8 MW from the turbine-generator, the plant can export 7.5 MW.

The wellhead pressure affects the flow rate of fluid from the well and the specific steam consumption for power production. To evaluate the effect of wellhead pressure on the specific steam consumption, Figure 16 has been constructed for different enthalpies of well water. The graph shows that the pressure which yields the highest power output per million pounds of water, i.e. the lowest specific steam consumption, increases as the enthalpy of well water increases.

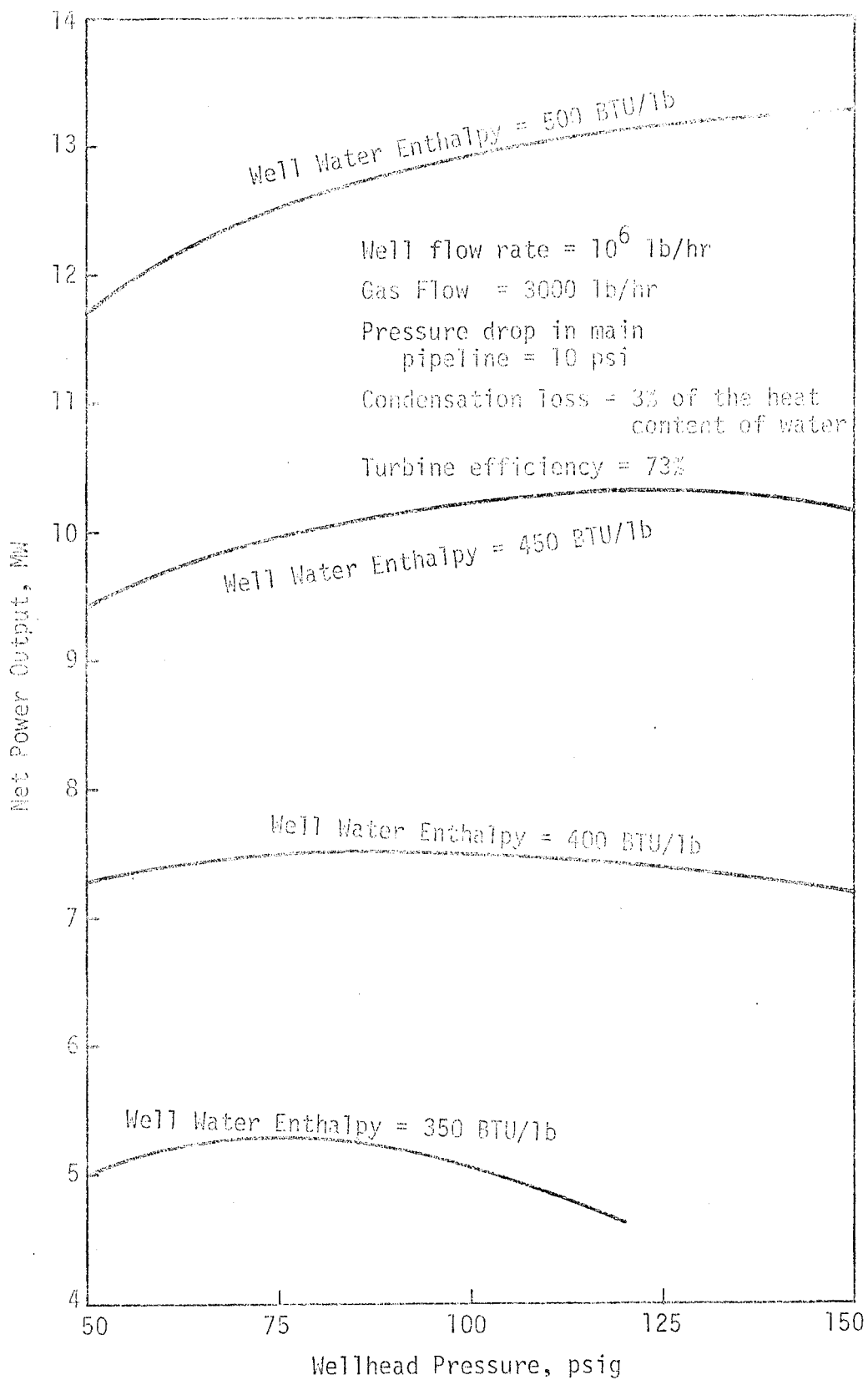


Figure 16. Variation of specific power output with wellhead pressure for a two-stage vapor flashing plant

E. Degree of Vacuum

The condensing pressures being reported in publications vary over a wide range: 1.5 in. Hg. abs. at Wairakei, 2 in. in Iceland, 2.5 in. at Larderello, 3.5 in. at Cerro Prieto, 3.75 in. at Matsukawa, and 4 in. at Geysers. The higher the vacuum to which a turbine can exhaust, the lower the specific steam consumption. However, the accessory power absorbed by the gas-extractor, cooling water pump, condensate pump and cooling tower fan increases as the condensing pressure decreases. The selection of the optimum pressure is influenced by such factors as the non-condensable gas content in the exhaust, and the temperature and the circulation rate of cooling water. To investigate the effects of condensing pressure, the net power output has been calculated with the basic assumptions given in the previous section, and plotted as a function of the condensing pressure and the non-condensable gas content in Figure 17. We notice that the higher the gas-content, the greater the steam consumption of the ejectors, and the higher the condensing pressure which yields maximum power output. In Figure 18, similar results are presented for the case of using a three-stage centrifugal compressor with intercoolers to substitute steam ejectors. The performance is based on the characteristic curves of Dal Secco [15]; the efficiency of the compressor is much lower than that of an ordinary air compressor.

The degree of vacuum in the condenser affects the capital cost and the unit cost of power. The performance of every component in the plant, particularly condenser and cooling tower, is influenced by the condensing pressure. Without detailed analyses of the costs, the optimum condensing pressure cannot be determined. James [30] conducted a study of the effect of gas-content on optimum condensing pressure. He concluded that the minimum generating cost of unit power is at the condensing pressure of about 5 in. Hg. abs. at which the capital cost is also the lowest; and if the gas content is higher than 25 percent in the vapor phase, a higher condensing pressure is required. The available information indicates that the condensing pressure of a geothermal plant should be much higher than that of a conventional power plant in the same locality, and a careful study of the costs is recommended before deciding upon the condensing pressure of a new plant.

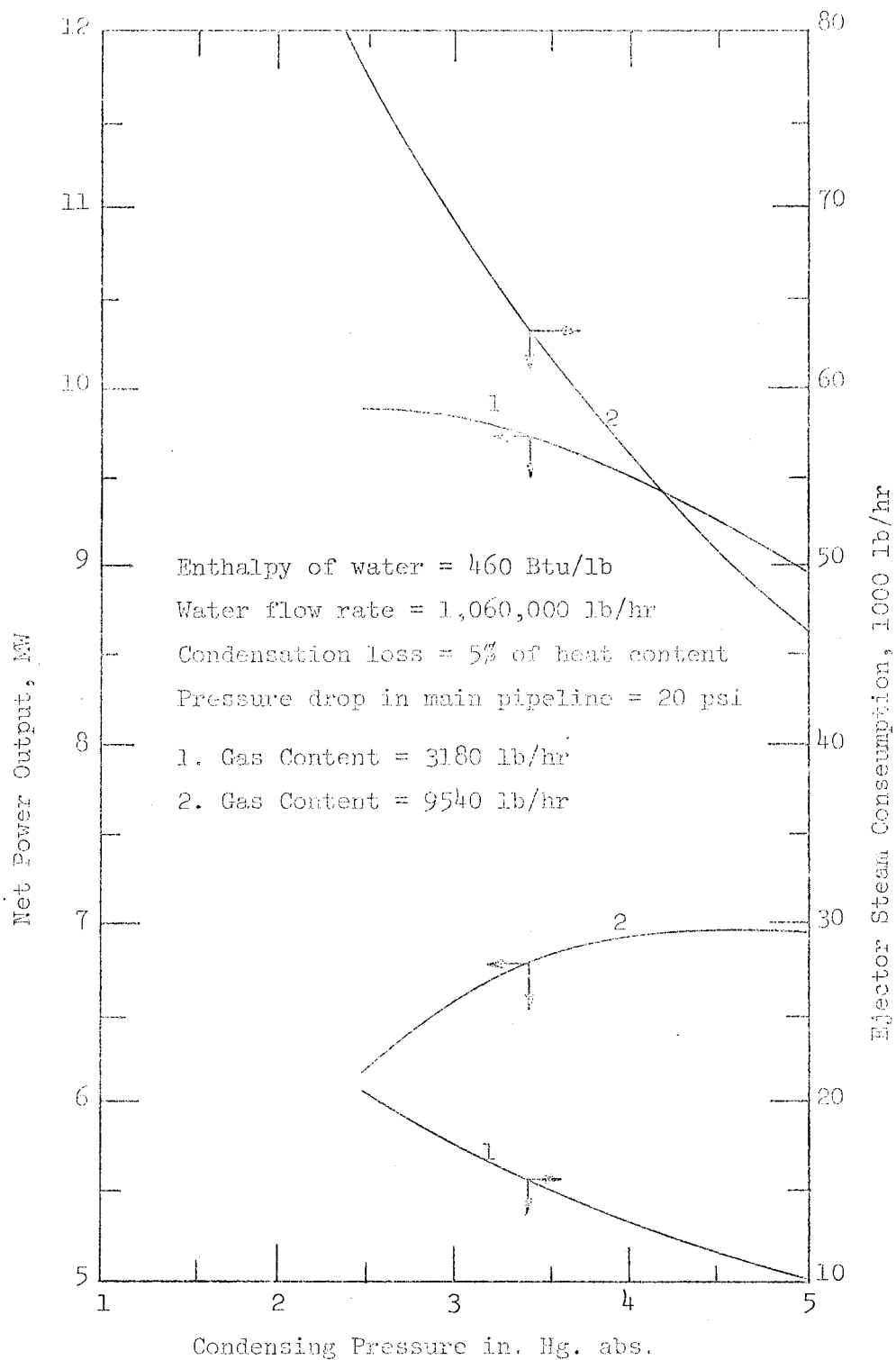


Figure 17. Power Output and Steam Consumption of Two-Stage Ejector versus Condensing Pressure.

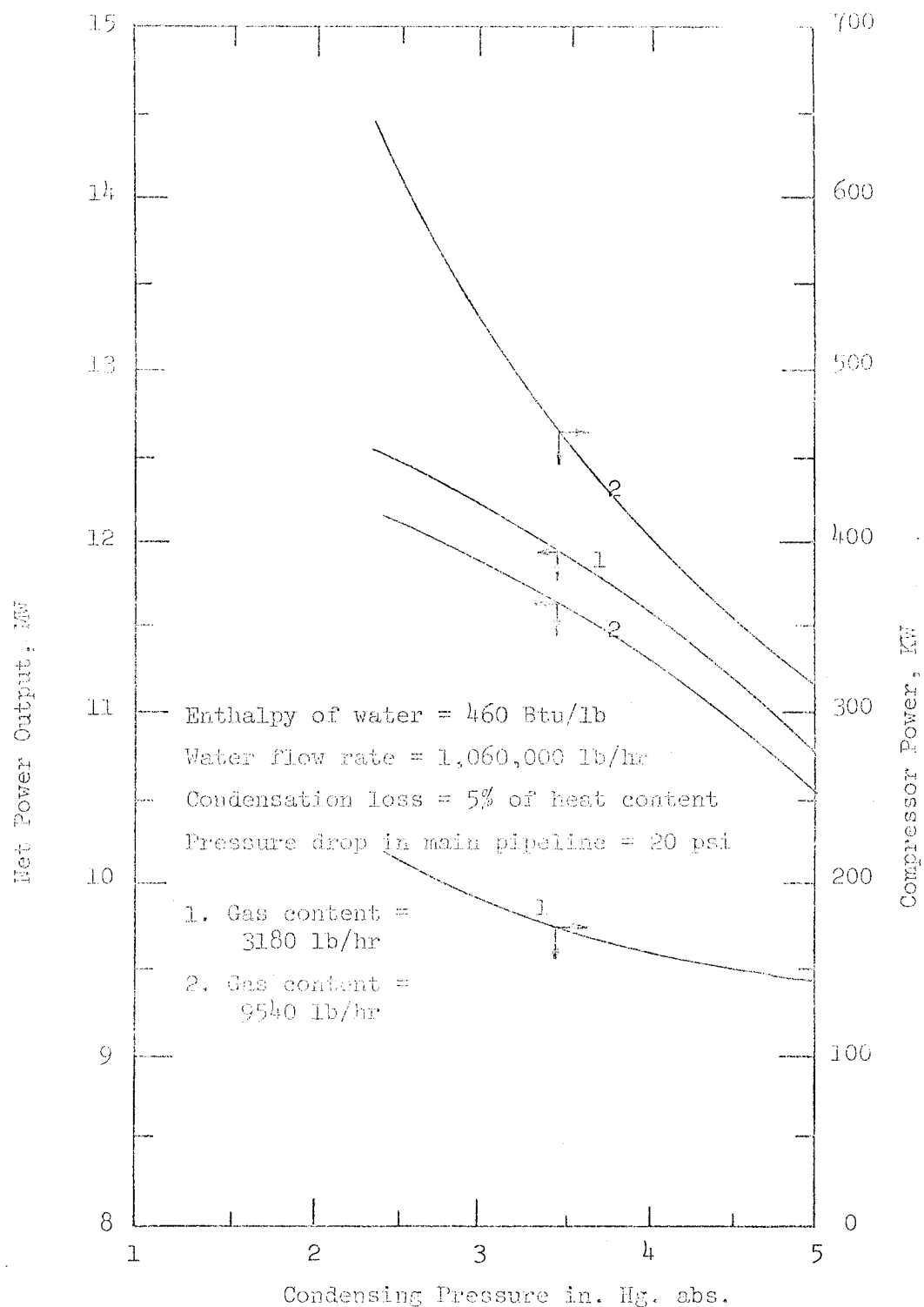


Figure 18. Power Output and Power Consumption of Three-Stage Compressor versus Condensing Pressure.

F. Cooling Tower

Cooling towers can be either dry or wet type. The dry cooling towers are not suitable for a condensing geothermal plant because: (1) they can cool water only to the limit of dry bulk temperature, (2) due to the mineral content of the condensate, the surfaces of the tubes are subject to fouling, (3) they are costly. The draft of a wet cooling tower can be produced mechanically or naturally. As compared with mechanical draft towers, natural draft towers occupy less space, consume less power and require less piping. They do not have many problems of fogging, drift and internal recirculation, and have long service life. However, they are not economical for plants having ratings less than 150 MW [41]. A mechanical draft tower can operate on forced or induced draft fan. The forced draft tower has the disadvantages of non-uniform air distribution and partial recirculation of vapor. As to induced draft towers, the crossflow design has distinct advantages over the counterflow arrangement. The distance of air travel in a crossflow tower is independent of the fill height; thus, both the distance of air travel and the height of the fill can be adjusted to minimize the loss of draft. In a counterflow tower, the distance of air travel varies directly with the fill height, which can be increased only at the expense of increasing the horsepower of the fan to compensate for the loss of draft. Cooling range is the temperature difference between the hot water entering the tower and the cold water leaving the tower. The crossflow tower is better than the counterflow tower to handle a large cooling range and a small approach, which is the difference between the wet bulb temperature of air and the temperature of leaving water. Besides, the pumping head of crossflow tower is lower than that of counterflow tower. In conclusion, an induced-draft crossflow tower is recommended for a geothermal power plant.

The performance of a cooling tower is a function of the cooling range, the approach and the ratio of water flow to air flow. The usual range of cooling towers in power plants is 10 to 40°F while the approach may be around 15°F. The approach has a great influence on the size of tower. The closer the cold water temperature approaches the wet bulb temperature, the greater the size of tower. The usual range of water-air mass flow ratio (L/G) is around 1.2 - 2.0 for mechanical draft towers in steam power plants.

In a cooling tower, there are numerous tiny water droplets, from which heat is transferred to the air in the free stream. The quantity of heat lost by the water is equal to the amount of heat gained by air; and the net transfer of energy may be expressed in terms of enthalpy potential as follows [6, 46]:

$$-L \text{ Cpw} = G \cdot dh = K \cdot a \cdot dV \cdot (h^1 - h) \quad (7.18)$$

or

$$-l \cdot (dx) (dy) \text{ Cpw} \cdot dt = g \cdot (dz)(dy) dh = K \cdot a \cdot dx \cdot dy \cdot dz \cdot (h^1 - h) \quad (7.19)$$

where dx , dy and dz are the dimensions of an infinitesimal volume dV , L and G are the flow rates of water and air in lb/hr respectively, l and g are the water and air flow rates per unit cross-sectional area perpendicular to the respective direction of flow in lb/hr/ft², $h^1 - h$ is the enthalpy potential defined as the difference between the enthalpy h^1 of saturated air at temperature t of water and the enthalpy of the unsaturated air in the free stream, a is the surface area of water per unit volume of tower in cu. ft./sq. ft., Cpw is the specific heat of water and K is the overall coefficient in lb/hr. sq. ft. which relates enthalpy potential to the heat transferred from water to air. For a small increment of tower with $\Delta x = \Delta z$, equation (7.19) may be revised as

$$\Delta h = \frac{-1}{g} \cdot \Delta t \quad (7.20)$$

$$\left(\frac{\Delta t}{h^1 - h} \right)_{\text{avg}} \cdot \left(1 \cdot \frac{\text{Btu}}{\text{lb} \cdot ^\circ\text{F}} \right) = \frac{-K \cdot a}{1} \cdot \Delta z \quad (7.21)$$

where Δz is the incremental height of tower in ft.

The dimensionless term in the right hand side of equation (7.21) is called number of transfer units (N T U). The dimensions of the tower can be related to NTU for a given design condition. To analyze a crossflow tower, the tower is divided into a series of incremental volumes, horizontally and vertically as shown in Figure 19. For the example, it is assumed: inlet water temperature 120°F, wet bulb temperature of air 75°F, ratio of water flow to air flow (l/g) 2.5 and NTU 0.2 for every increment. The calculations were started at the upper left corner, where the conditions of entering air and water are given. By trial and error, one finds in the first increment:

Water
 $t = 120^{\circ}\text{F}$
 $h^1 = 119.59 \text{ BTU/lb}$

AIR, Wet Bulb Temperature = 75°F
 $h = 38.61 \text{ Btu/lb}$

$\Delta t = 10.65$ $t = 109.25$ $h^1 = 90.91$ $h = 65.24$	$\Delta t = 7.1$ $t = 112.9$ $h^1 = 99.68$ $h = 82.39$	$\Delta t = 4.75$ $t = 115.71$ $h^1 = 105.71$ $h = 94.87$	$\Delta t = 3.2$ $t = 116.8$ $h^1 = 110.22$ $h = 102.37$	$\Delta t = 2.16$ $t = 117.84$ $h^1 = 113.24$ $h = 108.27$
$\Delta t = 7.16$ $t = 102.19$ $h^1 = 75.73$ $h = 55.51$	$\Delta t = 5.95$ $t = 106.95$ $h^1 = 85.63$ $h = 70.39$	$\Delta t = 4.77$ $t = 110.94$ $h^1 = 94.72$ $h = 82.32$	$\Delta t = 3.67$ $t = 113.15$ $h^1 = 100.31$ $h = 91.5$	$\Delta t = 2.82$ $t = 115.02$ $h^1 = 105.04$ $h = 98.55$
$\Delta t = 5.21$ $t = 96.98$ $h^1 = 66.52$ $h = 51.64$	$\Delta t = 4.66$ $t = 102.29$ $h^1 = 75.98$ $h = 63.29$	$\Delta t = 4.26$ $t = 106.68$ $h^1 = 85.04$ $h = 73.94$	$\Delta t = 3.51$ $t = 109.64$ $h^1 = 91.55$ $h = 82.72$	$\Delta t = 2.17$ $t = 112.85$ $h^1 = 99.55$ $h = 94.15$
$\Delta t = 3.98$ $t = 93$ $h^1 = 60.25$ $h = 48.56$	$\Delta t = 3.83$ $t = 98.46$ $h^1 = 69.03$ $h = 58.14$	$\Delta t = 3.68$ $t = 103$ $h^1 = 77.34$ $h = 67.34$	$\Delta t = 3.29$ $t = 106.35$ $h^1 = 84.31$ $h = 77.57$	$\Delta t = 3.19$ $t = 109.66$ $h^1 = 91.59$ $h = 83.55$
$\Delta t = 3.1$ $t = 89.9$ $h^1 = 55.79$ $h = 46.36$	$\Delta t = 3.2$ $t = 95.26$ $h^1 = 63.74$ $h = 54.36$	$\Delta t = 3.19$ $t = 99.81$ $h^1 = 71.39$ $h = 62.34$	$\Delta t = 3.01$ $t = 103.34$ $h^1 = 78.01$ $h = 69.87$	$\Delta t = 2.95$ $t = 106.71$ $h^1 = 85.10$ $h = 77.25$
$\Delta t = 2.49$ $t = 87.41$ $h^1 = 52.46$ $h = 44.84$	$\Delta t = 2.69$ $t = 92.57$ $h^1 = 59.62$ $h = 51.57$	$\Delta t = 2.78$ $t = 97.03$ $h^1 = 66.6$ $h = 58.52$	$\Delta t = 2.71$ $t = 100.63$ $h^1 = 72.88$ $h = 65.3$	$\Delta t = 2.71$ $t = 104$ $h^1 = 79.31$ $h = 72.08$
$\Delta t = 2.02$ $t = 85.39$ $h^1 = 49.91$ $h = 43.65$	$\Delta t = 2.3$ $t = 90.27$ $h^1 = 56.31$ $h = 49.39$	$\Delta t = 2.44$ $t = 94.59$ $h^1 = 62.68$ $h = 55.49$	$\Delta t = 2.57$ $t = 98.06$ $h^1 = 68.33$ $h = 61.02$	$\Delta t = 2.54$ $t = 101.46$ $h^1 = 74.41$ $h = 67.37$
$\Delta t = 1.64$ $t = 83.75$ $h^1 = 47.93$ $h = 42.72$	$\Delta t = 2.05$ $t = 88.22$ $h^1 = 53.52$ $h = 47.25$	$\Delta t = 2.21$ $t = 92.38$ $h^1 = 59.34$ $h = 52.78$	$\Delta t = 2.2$ $t = 95.86$ $h^1 = 64.7$ $h = 58.28$	$\Delta t = 2.26$ $t = 99.2$ $h^1 = 70.31$ $h = 63.93$
$\Delta t = 1.38$ $t = 82.38$ $h^1 = 46.33$ $h = 42.04$	$\Delta t = 1.67$ $t = 86.55$ $h^1 = 51.33$ $h = 46.22$	$\Delta t = 1.89$ $t = 90.49$ $h^1 = 56.62$ $h = 50.95$	$\Delta t = 1.96$ $t = 93.9$ $h^1 = 61.62$ $h = 55.95$	$\Delta t = 2.04$ $t = 97.16$ $h^1 = 66.82$ $h = 60.95$

Figure 19. Distribution of Water Temperature and Air Enthalpy
in a Cross-Flow Cooling Tower

$(h^1 - h)_{in} = 80.98 \text{ Btu/lb}$, $(h^1 - h)_{avg.} = 53.23 \text{ Btu/lb}$, $\Delta t = 10.65^\circ\text{F}$,
 $\Delta h = 26.63 \text{ Btu/lb}$, $(h^1 - h)_{out} = 24.97 \text{ Btu/lb}$, and temperature of leaving
 water = 109.35°F . The calculations proceed progressively to the subsequent
 increments. There are five horizontal and nine vertical increments. The
 ratio of water flow in lb/hr to air flow in lb/hr is $(1/g) \cdot (5/9) = 1.39$.

The height and length of the tower are related to the total number of
 transfer units, designated as number of vertical transfer units (NVTU) in the
 vertical direction and number of horizontal transfer units (NHTU) in the
 horizontal direction. In Figure 19, $NVTU = (\text{no. of vertical increments})$
 $(HTU) = (9)(0.2) = 1.8$, and $NHTU = (5)(0.2) = 1.0$. Since equations (7.20) and
 (7.21) are based on the condition that the height of an increment is equal to
 its width, $\Delta x = \Delta z$, the ratio $NVTU/NHTU$ gives the proportion of the height to
 the width of tower.

The resulting average temperature of water is 90.1°F for the example
 given in Figure 19. The variations of water temperature with different values
 of NHTU and NVTU for the typical example were calculated and are given in
 Table 3. To facilitate the application of the calculated results to the
 estimation of tower performance, Figure 20 was constructed with water temperature
 as ordinate, and NHTU and NVTU as the other two variable parameters. For a
 given value of leaving temperature of water, there are various combinations of
 the L/G ratio, the number of horizontal increments and the number of vertical
 increments of tower. The volume of tower fill and the draft loss are affected
 by the height and the width. The recommended design point is at L/G ratio of
 about 1.6 [37].

The values of $K \cdot a$ have been experimentally determined in relation to air
 flow (g) for a splash filled tower by Snyder [39], as shown in Figure 21. At
 $g = 1000 \text{ lb/hr} - \text{ft}^2$, the value of $K \cdot a$ is $93 \text{ lb/hr} - \text{ft}^3$ from the diagram.
 Since the value of NVTU is equal to the product of the number of vertical
 increments and NTU, by equation (7.21) we have: tower height = $(NVTU) (1/K \cdot a)$;
 likewise tower width = $(NHTU)(1/K \cdot a)$. For the L/G ratio of 1.39, one finds
 $NHTU = 1.0$ and $NVTU = 1.8$ from Figure 19; thus tower height = 45.9 ft. and
 tower width = 25.5 ft. Usually the air enters a tower from two opposite sides.
 If so, the width of tower becomes double, $2 \times 25.5 = 51 \text{ ft}$. For 24,000 gpm of
 water flow (L), the dimensions of the tower are 48.4 ft height \times 53.6 ft wide \times
 96 ft long. If the tower is divided into three cells, the length of each cell
 is 32 ft.

Table 3. Temperatures of Water Leaving a Cross-Flow Cooling Tower

NHTU	HVTU								
	.2	.4	.6	.8	1.0	1.2	1.4	1.6	1.8
.2	109.35°F	102.19	96.98	93	89.9	87.41	85.39	83.75	82.38
.4	111.13	104.57	99.64	95.73	92.58	89.99	87.63	85.99	84.47
.6	112.65	106.69	101.98	98.15	94.94	92.34	90.08	88.12	86.47
.8	113.69	108.31	103.90	100.20	97.06	94.41	92.06	90.05	88.33
1.0	114.52	109.65	105.69	102.09	99	96.33	93.95	91.88	90.10

Flow Ratio $l/g = 2.5$

Wet Bulb Temperature = 75°F

Temperature of Leaving Water = 120°F

NTU = 0.2

NHTU = (No. of horizontal increments) (NTU)

NVTU = (No. of vertical increments) (NTU)

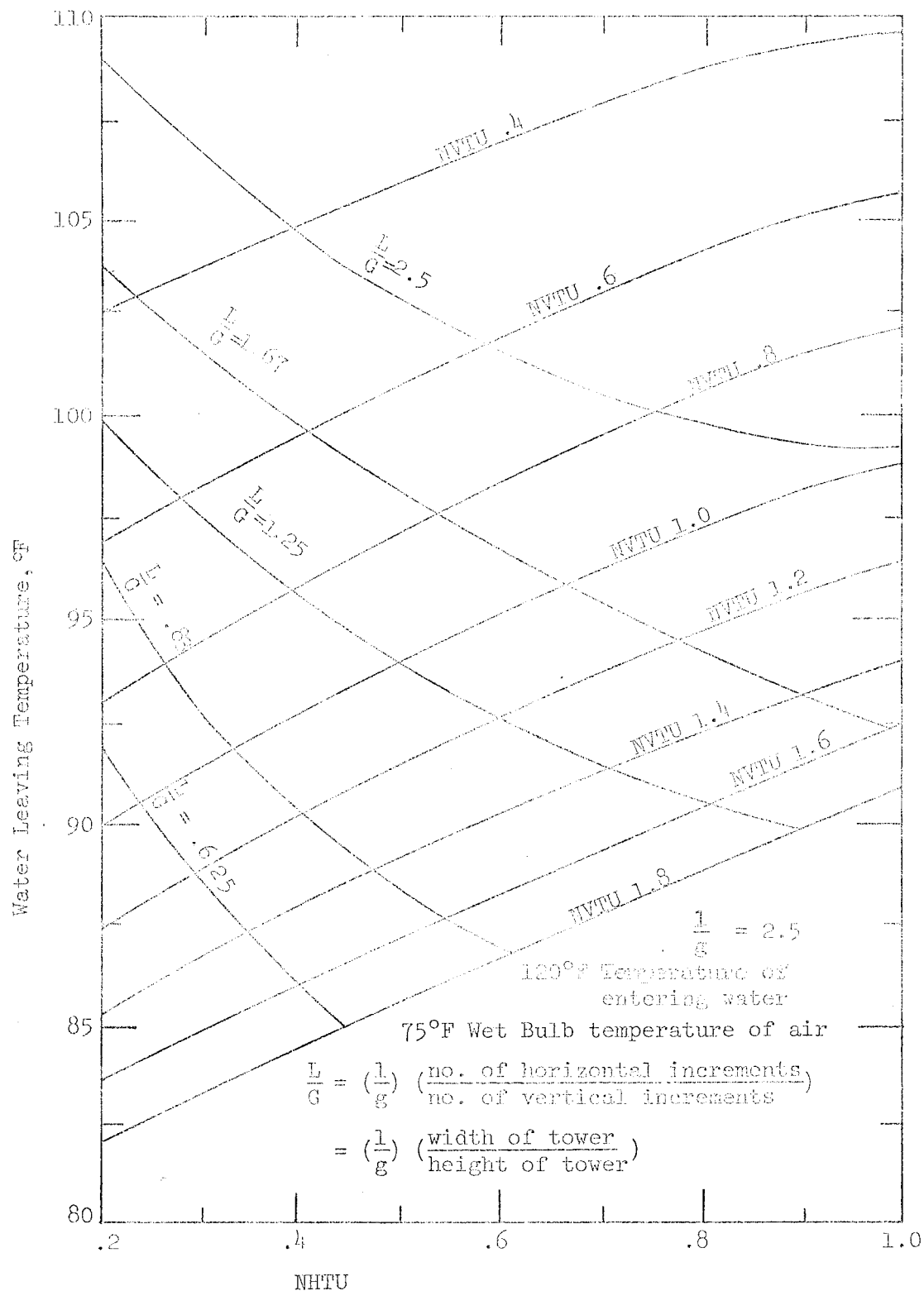


Figure 20. Performance of a Cross-Flow Cooling Tower

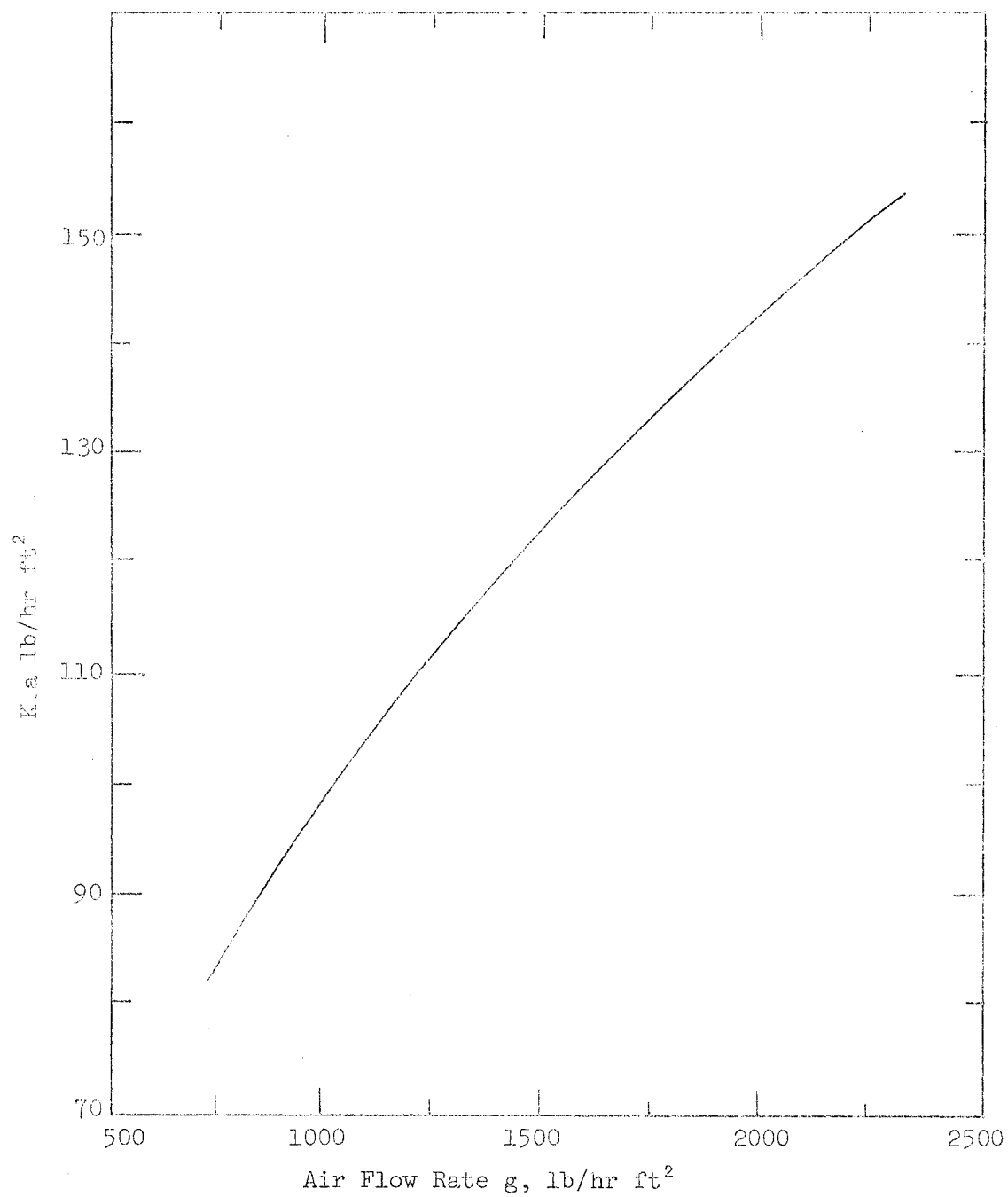


Figure 21. Tower Coefficient $K'a$ versus Air Flow Rate g [39]

Special materials are needed to resist the chemical reaction of the impurities in water. The condensate and cooling tower pipes can be made of stainless steel, aluminum or plastic coated material. The casing is usually made of asbestos cement. For additional resistance to corrosion, the wetted surface may be coated with tar, epoxy compounds or synthetic rubbers. The drift eliminators can be of treated woods, galvanized steel, stainless steel, plastic or coated asbestos. The material of tower filling can be redwood, plastics, aluminum, stainless steel or ceramics. The last three are expensive. Test results at Geysers showed polyvinyl chloride as a favourable material for the tower fill.

G. Arrangement of Equipment

Several arrangements of the power plant are possible, depending upon the topography of the site which can be hilly or flat. Saporiti [36] considered the arrangements of plants and found that 30 to 50 ft. difference in level is desirable. The plant arrangements can be classified into two categories: with underlying condensers and with overlying condensers.

In plants with underlying condensers, the condenser is placed directly beneath the turbine to make full use of the vacuum. Either the turbine is placed on an elevated level, or a deep excavation is necessary for the accommodation of the hot well. At Larderello, some of the turbines are placed at the ground level with the hot-wells in the excavated pits. The barometric pipes are lowered into the hot-wells till they reach the underground vessels which are connected to underground tunnels for the discharge of overflow. The condensate pumps push the hot water through the spray nozzles of the cooling tower. If the pump fails, the excess water is discharged to the tunnel to avoid the danger of overflow into the steam pipe. The major drawback of this scheme is the high costs of excavation and tunnels.

To reduce the depth of excavated pit and to remove the necessity of laying underground tunnels, a modified arrangement is presented in Figure 22. The condenser is divided into two sections for the reduction of its height. To do away with the discharge tunnels, the level controls and interlocks are provided to avoid possible overflow in the case of sudden shutdown of the circulating water pump. If the pump fails, the interlock mechanism trips the turbine admission valve, breaks the vacuum in the condenser through a vacuum breaker

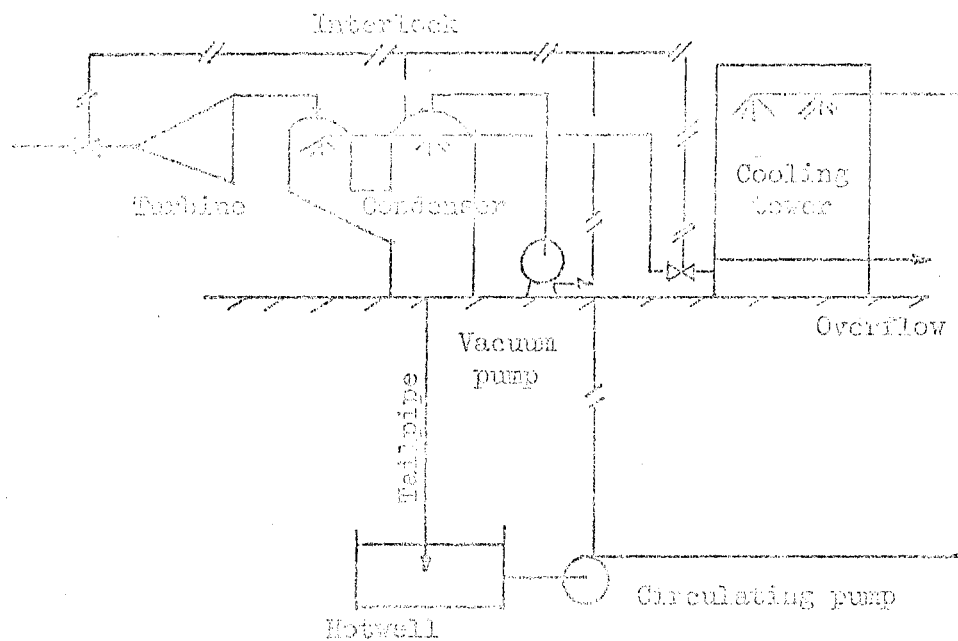


Figure 22. Plant Arrangement with Underlying Condenser

and closes the quick-closing valve at the inlet of cooling water line. The level of cooling water should be slightly lower than that of the condenser spray nozzles so that the cooling water will not flow into the condenser after the vacuum breaker is released.

Plant arrangements with overlying condensers, as being adopted in some of the plants at Geysers, do not require excavation work. The barometric condenser must be installed above the turbine level. Due to the pressure drop in the exhaust duct, the back pressure of turbine increases. Figures 23, 24 and 25 illustrate three arrangements with overlying condensers: with upper cooling towers, with lower cooling towers and with cooling towers at the turbine level. If the turbine, hot well and cooling tower are located at the same level, then two circulating water pumps are required. If there is 30 to 50 ft. difference in the ground level, the plant arrangement shown in Figures 23 or 25 may be adopted; one of the advantages is that only one circulating pump is required which simplifies controls and safety devices.

To avoid the high costs of the supporting structure of the condenser and the long turbine exhaust duct, a low-level type condenser may be used; however, a failure of the circulating pump can cause serious damage to the turbine, and the pumping power is very high. Since the barometric pipe, the cooling water-inlet pipe and the non-condensable gas extraction pipe are nearly at the same temperature, they can be used together to support the condenser for reducing the cost of supporting structure.

If the site for power station is on a steep slope, the setup as shown in Figure 22 is an ideal arrangement with the tailpipe of the main condenser to follow the natural contour. It gives the advantage of low back pressure without incurring the cost of excavation, and only one circulating pump is required.

CHAPTER VIII.

PLANT SITING

The site selection of a geothermal power plant is rather inflexible and primarily depends upon the location of producing reservoirs. However, there is some flexibility in selecting a site for a very large reservoir, which may cover several thousand acres of land. The process of selection involves the

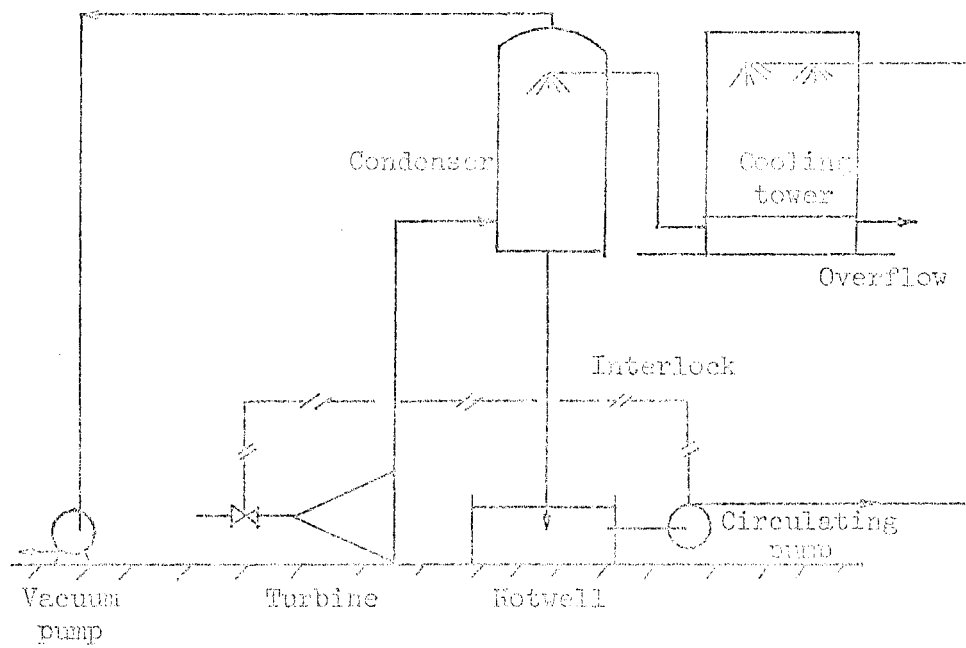


Figure 23. Plant Arrangement with Overlying Condenser and Upper Cooling Tower

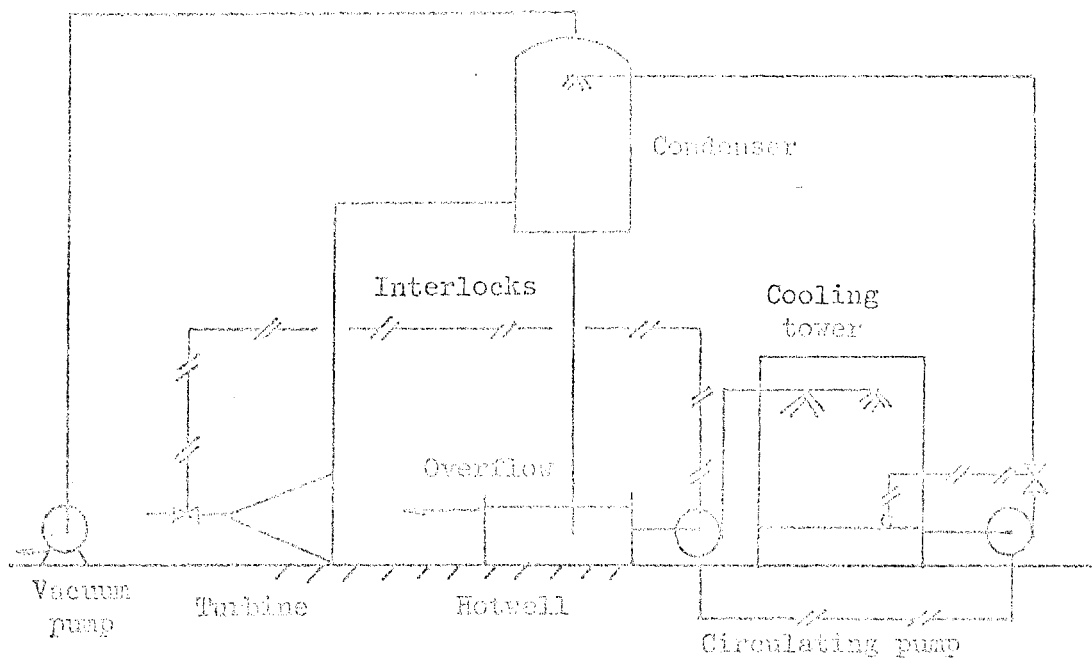


Figure 24. Plant Arrangement with Overlying Condenser
for Cooling Tower at Turbine Level

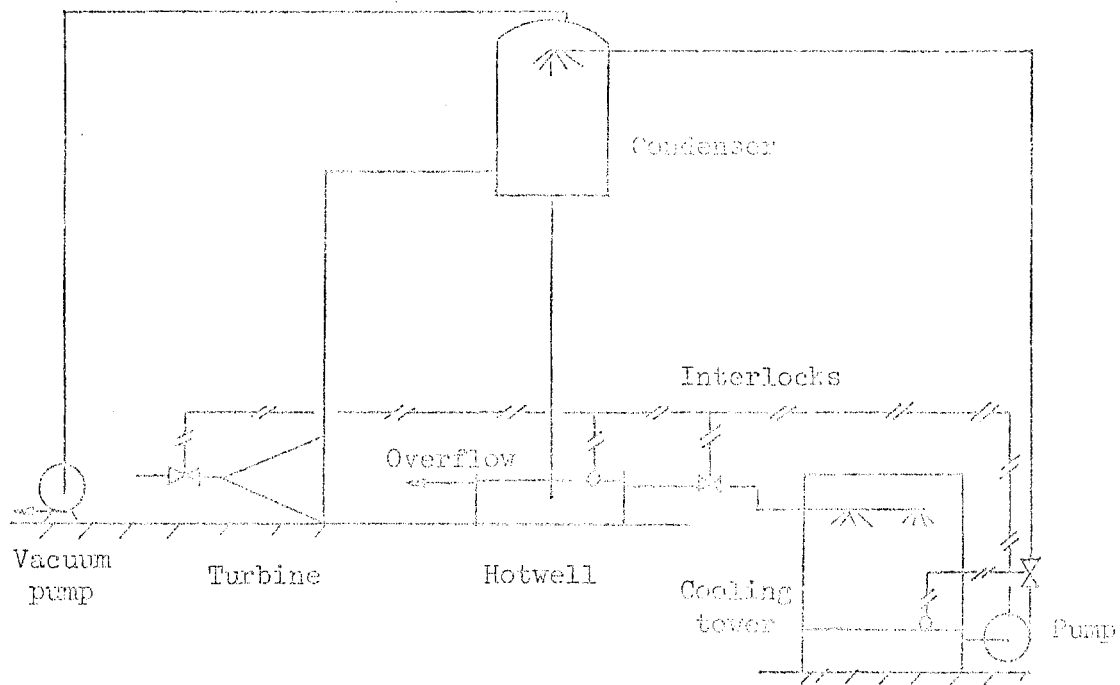


Figure 25. Plant Arrangement with Overlying Condenser and Lower Cooling Tower

identification of the factors affecting the construction and operation, the study of the constraints imposed by the requirements of plant design, the field investigation of all the possible sites and the value judgment. The major elements that contribute to bring about the decision of final selection are: availability of energy source, seismology, subsurface hydrology, land use law, population distribution, location and size of load center of utilities, electrical transmission network, ecology, environment, politics, historical and aesthetic considerations and costs of plant.

The wellhead elevation of production wells should be close to the ground-water table. The cost of well increases with the increase of depth while the flow rate decreases. The distance between individual production wells may be 500 to 1,000 ft. and the power station should be designed and constructed at or near the wellhead of one or more production wells. If the discharge of hot waste brine into a stream or ocean is prohibited, the waste may be reinjected into the reservoir through a reinjection well located within one mile of the power station. The depth and location of the reinjection well should be carefully determined, so the nearby groundwater for domestic or individual uses will not be contaminated. The buildings of the station should not be located within one-quarter mile of a surface fault or near the epicenters of previous earthquakes.

Patterns of land use, existing utility facilities, population projections, patterns of economic growth and local politics all affect the selection of a preferred site. No site can satisfy all the requirements, and the final selection has to be based on the compromise between the favorable and unfavorable characteristics of each site after the definitely unsuitable sites are eliminated. Ecological and environmental impacts must be considered in site selection. To avoid damage caused by gaseous effluents, the patterns of air dispersion must be studied with respect to the topograph and seasonal change of climate. On an island, the preferred site is on the leeward side where the tradewind can sweep the undesirable gases and fumes into the ocean. The potential site must meet the requirements of the local zoning codes and the acceptance of the general public. Any possible sites near a National Park and Forest, wildlife refuge, scenic or historical area cannot be accepted favorably by the public.

Unlike the conventional power plants, which operate in a closed cycle, the geothermal power plants do not need make-up water for their cooling towers. Adequate water supply for plant operation is not a requirement for siting a geothermal project, although there is need for water during the drilling and construction. In fact, in an arid area it could be economical to desalt a part of the waste brine for producing fresh water as a by-product of the geothermal plant.

In consideration of the transmission cost, it is desirable to locate the plant not far from the load center of power distribution. The acquisition of rights-of-way across public land is becoming increasingly difficult because of the public awareness of the beauties of environment. In addition, a long transmission line is too costly.

From the viewpoint of station designer, the turbine should be set on a firm foundation. A site having non-fractured bedrock can substantially reduce the cost of foundation. Practically all the condensing plants consist of barometric condensers and cooling towers. A 30-ft high cliff at the site can simplify the arrangement of the tail-pipe of a barometric condenser and improve the operation of the plant. The exhaust air from the cooling tower is approximately saturated, and fogging may occur when the exhaust air mixes with the humid ambient air. If the tower is too close to a highway, the impaired visibility due to fog can lead to traffic accidents. For a 10 MW plant, the land area required by the station is about one acre. The capital cost of the station largely depends upon the nature of geothermal fluid and the type of plant; however, the siting is also one of the cost factors.

CHAPTER IX.

CONCLUSIONS

The mixture of steam and water can be transported in a single pipe without the problems of water hammer, cavitation and vibration. The pressure drop of two-phase flow may be determined by using Lockhart and Martinelli correlation described in Chapter II. After the mixture reaches a plant, the steam can be separated from the water in a cyclone separator and the water enters a

subsequent cyclone separator to produce low-pressure flash steam. Both the power output and the quality of exhaust steam increase with the number of flashing stages. The steam production from each stage and the available energy per pound of steam depend upon the operating pressure of the stage. The lower the flashing pressure, the higher the steam production rate, but the available energy drops as the pressure decreases. The power output from the turbine is proportional to the product of steam flow rate and available energy. For obtaining the maximum power, approximately the optimum flashing pressures should correspond to the saturation temperatures which are equally divided between the saturation temperature of hot water to the plant and the condensing temperature. The calculated results by numerical method in Chapter III confirm the validity of the approximate rate of optimum flashing pressures.

The wellhead pressure is closely related to the reservoir temperature, the flow rate characteristics of the well, the reservoir life and the power productivity. In Chapter IV, the effect of wellhead pressure and number of flashing stages on power output has been analyzed. Although the power production increases with the number of flashing stages, the number of flashing stages may have to be restricted to two for economical reasons. For a typical well, the flow rate decreases as the wellhead pressure increases. The optimum flashing pressure for a two-stage flashing plant is around 100 psig. The curve of power output versus pressure in Figure 4 is flat near the points of maximum power output. It implies that there is some leeway for selecting wellhead pressure.

The bottom-outlet-cyclone separators with geometric proportions as described in Figure 5 are capable of producing steam at a quality higher than 99.9 percent. They have been tested and adopted in New Zealand. From the rate of mass flow, inlet steam wetness and separating pressure, the inlet velocity can be determined from graphs of Figures 6 and 7 for the given values of pressure drop and separating efficiency. From the inlet velocity, the inlet diameter D and hence the cyclone dimensions can be determined according to the proportions in Figure 5.

The most critical component is the turbine, of which the materials must be carefully selected. There is no standard for turbines using geothermal steam. Each turbine is specially designed to suit the chemical quality of the steam and other operating conditions. The exact performance of a turbine is

not known until the specification from the manufacturer is received. For the purpose of a heat balance, there is sufficient information from the publication to estimate the steam rates in terms of the size, loading, throttle pressure, inlet temperature and condensing pressure.

Steam jet ejectors should be used when the non-condensable gas content is low. For higher content, the use of centrifugal compressors is advantageous. For much higher gas content, it may be desirable to use a back pressure turbine, and thus the heat rejection equipment is eliminated completely. Since there is no need to recover the condensate, it is permissible to mix the cooling water with the condensate. As a rule, condensers of direct contact barometric type are used in vapor-flashing geothermal plants. Surface condensers are not suitable because of their high initial costs, poor heat transfer coefficients and large sizes. An induced draft crossflow cooling tower is more suitable than other types of cooling towers, and the procedures of estimating the sizes are presented in Chapter VII with the help of an illustrative example.

The condensing pressure influences the capital cost of the plant and the unit cost of power produced. The lower the condensing pressure to which a turbine exhausts, the lower the steam rate of turbine. However, the accessory power required by gas-extractors, circulating water pump and cooling tower fan increases as the condensing pressure decreases. Figure 13 illustrates the result of a sample heat balance, by which the optimum condensing pressure of a plant may be determined. Figures 17 and 18 show the effects of non-condensable gas content and condensing pressure on the net power output. The higher the gas content, the higher the optimum condensing pressure for the maximum power output. The final selection of condensing pressure necessitates the cost evaluation of the plant.

The arrangement of heat rejection equipment depends primarily on the topography of the site. A 30-ft cliff at the site is very desirable for simplifying the plant arrangement if the plant includes a barometric condenser. A discussion on arrangements in relation to the floor level is given in Section F of Chapter VII.

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